Evaluation of Grundfos CRE 15-3 Variable Speed Centrifugal Pump and Worthington Constant Speed Centrifugal Pump Applications in KU Steam Power Plant

By

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Evaluation of Grundfos CRE 15-3 Variable Speed Centrifugal Pump and Worthington Constant Speed Centrifugal Pump Applications in KU Steam Power Plant

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Abstract

The applicability of Variable Frequency Drives (VFD) has been widely used to substantially contribute to energy consumption savings. These savings have been achieved for a relatively low static head, especially when the flow rate demand or required discharge head is not constant and varies often with time. For the project described herein, a comparative study was conducted between a pair of Grundfos pumps and a Worthington pump in the steam power plant at the University of Kansas.

These two types of pumps are used to supply condensate water to two different components of the steam power plant. One component is the deaerator tank, whose function is to preheat and deaerate the condensate water, before it is supplied to the boiler. This deareator tank is located in the basement of the steam power plant. The second component is the vent condensing heat exchanger, which is located on the first floor of the steam power plant. The vent condenser is used to reclaim some of the energy in the escaping non-condensable gases. All of the pumps' operating data, such as discharge pressure, flow rate, and power consumption, were recorded for both of the pumps for two cases.

In Case 1, the Grundfos pumps were operated in constant discharge pressure mode, and they were able to supply condensate water to both the deaerator tank and the vent condenser. The Worthington pump operated normally and was able to supply water to both components. However, in Case 1, it was found that the Grundfos pumps consumed 2.86 kW more power and provide a higher discharge pressure (1.97 PSIG more) and flow rate (52.534 GPM more) than the Worthington pump. The Grundfos pumps can operate in either pressure control mode or level control mode.

In Case 2, the Grundfos pumps ran in level control mode; and they were only able to feed the deaerator tank. They were not able to feed the vent condenser due to the vent condenser's high static head (40 ft). For Case 2, the same task (feed the deaerator, but not the vent condenser) was assigned to the Worthington pump so that the two pump types had comparable jobs. In Case 2, Test #2, the Grundfos pumps consumed 2.83 kW less power than the Worthington pump. Moreover, the Grundfos pumps provided a discharge pressure and flow rate that was 18.684 GPM lower and 32.43 PSIG lower, respectively, than those of the Worthington pump.

For Case 2, a life cycle cost analysis was performed in order to compare both types of the pumps' total life cycle costs, and to determine which of the pump types had the lowest total life cycle costs. The present value of all LCCs for 20 years was $472,358 for the Grundfos pumps when running in level control mode, and the present value for the Worthington pump was $497,776. Thus, the net savings from running the Grundfos pumps in level control mode for 20 years, including all costs, was $25,418. However, in Case 2, the Grundfos pumps were incapable of feeding the vent condenser with condensate water. The vent condenser's main purpose is to capture some of the energy from the non-condensable gases that are removed from the deaerator tank. The savings from using the vent condenser was found to be $812,547 over the same 20 year life. This savings was obvious more than the savings from using the Grundfos pumps when they operated in level control mode. Due to the fact that the savings from using the vent condenser was much more than the savings from using the Grundfos pumps, it is highly recommended to install another vent condenser so that some of the remaining non-condensable gases' energy can be captured. More preferably, if these two vent condensers were installed next to the DA tank in the basement of the steam power plant, the steam power plant could save even more by using Grundfos pumps and vent condensers.

To My Angel, My Wife

To My Beloved Son, Jafar

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Nomenclature

A = Pipe internal cross-sectional area (ft^2)

A_o = Series of equal cash amounts ($)

A_t = Annually recurring cash amounts ($)

bhp = Brake pump horsepower (HP)

BD = Blowdown (GPM)

BEP = Best Efficiency Point (%)

c_{pwater} = Specific heat of water ($Btu/lb_m{}^oF$)

C_1 = Units conversion factor (1714($\frac{GPM\ lb}{in^2\ hp}$))

C_c = Contraction coefficient (-)

C_d = Decommissioning/disposal costs (including restoration of the local environment and disposal of auxiliary services) ($)

C_e = Energy costs (predicted cost for system operation, including pump driver, controls, and any auxiliary services) ($)

C_{env} = Environmental costs (contamination from pumped liquid and auxiliary equipment) ($)

C_{ic} = Initial costs, purchase price (pump, system, pipe, auxiliary services) ($)

C_{in} = Installation and commissioning cost (including training) ($)

C_m = Maintenance and repair costs (routine and predicted repairs) ($)

C_o = Operational costs (labor cost of normal system supervision) ($)

C_s = Down time costs (loss of production) ($)

C_V = Flow coefficient $\left(\mathrm{GPM}\sqrt{\frac{\mathrm{in}^2}{\mathrm{lb}}}\right)$

CR = Condensate return (GPM)

d = Discount rate (%)

$d^{'}$ = Pipe diameter after/before contraction/expansion (in)

D = Inside pipe diameter (ft)

DN = Diameter nominal (in or mm)

e = Constant escalation rate provide by DOE (%)

E = Present-value energy costs ($)

E_{BFW} = Boiler feed water energy (Btu)

E_{fuel} = Fuel energy (Btu)

E_p = Pump energy consumption (kWh)

E_{steam} = Generated steam energy (Btu)

E_{vent} = Energy recovered from the vent condenser (Btu)

f = Friction factor

F_t = Future cash amount in PV formula for one time amounts ($)

FEMP UPV* = Factor for use with energy costs (-)

FW= Feed water (GPM)

g = Sea level acceleration due to gravity (32.174 ft/s^2)

$h_{cond\ in}$ = Condensate water enthalpy entering the vent condenser (Btu/lb)

$h_{cond\ out}$ = Condensate water enthalpy exiting the vent condenser (Btu/lb)

h_f = Friction head (ft)

h_{minor} = Minor head losses (ft)

$h_{non\text{-}cond.\ gases\text{-}in}$ = Non-condensable gases enthalpy entering the vent condenser (Btu/lb)

$h_{non\text{-}cond.\ gases\text{-}out}$ = Non-condensable gases enthalpy existing the vent condenser (Btu/lb)

$h_{sat.steam}$ = Saturated steam enthalpy generated from boiler (Btu/lb)

$h_{sat.water\ (BFW)}$ = Boiler feed water enthalpy (Btu/lb)

H = Total pressure head of water (ft)

H_{bep} = Pump head at best efficiency point (ft)

H_d = Total water pressure head at pump discharge (ft)

H_{duty} = Pump head at duty point (ft)

$H_{pump\ total}$ = Total pressure head developed by pump (ft)

H_s = Total water pressure head at the pump suction (ft)

I = Present-value investment costs ($)

K = Local flow resistance coefficient (-)

L = Pipe length (ft)

LCC = Life cycle cost ($)

LHV_{fuel} = Natural gas Lower Heating Value (Btu/ft^3)

m^o_{BWF} = Boiler feedwater flow rate (lb_m/hr)

m^o_{fuel} = Fuel flow rate (ft^3/hr)

m^o_{steam} = Steam flow rate (lb_m/hr)

m^o_{water} = Liquid water flow rate (lb_m/hr)

Mu = Make-up water flow rate (GPM)

n = Pump impeller rotational speed (rpm)

n_{max} = Maximum speed of variable speed pump (rpm)

ni = Number of years in the study period (years)

NPS = Nominal pipe size (in)

OM&R = Present-value non-fuel operating, maintenance, and repair costs ($)

P = Total gauge pressure (PSIG)

P_d = Pressure at a pump discharge side (PSIG)

P_s = Pressure at a pump suction side (PSIG)

PV = Present Value ($)

Q = Volumetric flow rate (GPM)

Q_{bep} = Pump flow rate at best efficiency point (GPM)

Q_{duty} = Pump flow rate at duty point (GPM)

Q_T = Total moved water volume (Gallons)

Q_v = Vent condenser flow rate (GPM)

R = Reynold's number (-)

Repl = Present-value capital replacement costs ($)

Res = Present-value of residual value (resale value, scrap value, salvage value) less disposal costs ($)

S = Steam flow rate (GPM)

S_G = Steam generated (lb)

t = Time (years, months, days, hours, minutes)

t_o, t_1, t_2, t_j, t_{j-1} = The 0^{th} , 1^{st}, $j\text{-}1^{th}$, j^{th} time points for trapezoidal integration (sec)

T_i = Integral time of the variable speed pumps' controller (sec)

UPV = Uniform present value factor (-)

UPV* = Uniform present value factor modified for price escalation (-)

V_d = Discharge pipe water velocity (ft/s)

V_s = Suction pipe water velocity (ft/s)

$\dot{W}$ = Pump power (kW or HP)

W = Present-value of water costs ($)

WL = DA tank water level (%)

X1 = Recorded power consumption obtained from curve fittings (kW)

X2 = Power consumption from the pump curve (kW)

X3 = Power consumption obtained from Eq. (18) (kW)

Y = Number of calibration readings (-)

Z = Elevation (ft)

Z_d = Elevation above (+) or below (-) datum plane on discharge side of pump (ft)

Z_s = Elevation above (+) or below (-) datum plane on suction side of pump (ft)

Greek

α = Pump speed ratio (%)

γ = Specific weight of water (lb/ft^3)

γ' = Specific gravity (-)

γ_d = Specific weight of water at a pump discharge side (lb/ft^3)

γ_s = Specific weight of water at a pump suction side (lb/ft^3)

ΔP = Differential developed pressure across a pump (PSIG)

ΔT_{rise} = Difference between water inlet and exit temperatures (^oF)

ε = Pipe roughness (in)

η = Efficiency (-)

η_{boiler} = Boiler efficiency (-)

ν = Kinematic viscosity (ft^2/s)

ρ_{water} = Water density (lb/ft^3)

Subscripts

1, 2 = Different pump states/speeds

A, B = Two different points in a pipe

ft = Fuel type

m = Motor

p = Pump

reg = Region

rt = Rate type

v = Variable frequency drive

vent = Vent condenser

non-cond = Non-condensable gases

Abbreviations

BFWP = Boiler Feed Water Pump

BLCC5 = Building Life Cycle Cost Software (from DOE)

BWF = Boiler Feed Water

CSP = Constant Speed Pump

DOE = U.S. Department of Energy

FEMP = Federal Energy Management Program

LCCA = Life Cycle Cost Analysis

SPV = Single Present Value Factor

VFD = Variable Frequency Drive

VSD = Variable Speed Drive

VSP = Variable Speed Pump

Chapter 1

1.0 Objectives

This Project has been developed in order to investigate the most cost effective components that provide the highest savings in the steam power plant located at the University of Kansas, Lawrence campus. Both the vent condensing heat exchanger and the Grundfos pumps' savings were investigated. The Grundfos pumps can save energy when running in level control mode. In level control mode, they were incapable of feeding condensate water to the vent condenser located on the first floor of the steam power plant. In pressure control mode, they could provide vent condenser water. Hence, it was necessary to investigate the vent condenser's savings. The vent condenser's main purpose is to capture some of the energy from the non-condensable gases escaping from the DA tank. As a result, this project used LCCA in order to compare the savings from the Grundfos pumps when running in level control mode and from the vent condenser.

1.1 Pumps

A centrifugal pump is one of the components of HVAC systems. Centrifugal pumps have different kinds of jobs from providing water to a boiler to moving chilled and hot water to utilities equipment so that heat rejection takes place in the chillers [1]. Modern technology has added a few features to enhance pump performance. However, many factors, such as market demand and cost-effective building construction, have forced manufacturers to produce economical, reliable, and long-term service pumps. There are some important points that have to be considered when choosing a pump to perform a specific job. Pump efficiency is one of the important factors, since pump efficiency is defined as the percentage of supplied energy that is converted to useful work. This work is presented in the form of discharge pressure and flow rate. The reason for not having a full conversion of energy is the losses. They can be sorted into different categories: (a) drive-system or motor losses due to bearing friction besides motor efficiency losses, (b) mechanical losses, for instance shaft bearing friction and drag forces on the impeller, (c) water circulation due to clearance between impeller and volute, i.e., the enclosed casing of the pump in which the water is moved by the impeller, and (d) hydraulic losses due to converting mechanical work to fluid velocity [1].

Selecting the right pump for a specific system requires knowing the head capacity curve for that system (i.e., the system curve) [2]. This is the first step in designing a pumping system. Computing the system loss curve is typically based on calculating the losses associated with pipes and fittings [2]. However, calculating these losses for pipes which are more than ten years old may not be as accurate as the same calculations for new pipes, due to the fact that the pipe roughness changes with time. The pipe roughness (being determined theoretically) might predict a lower or higher friction head for a pumping system. This may lead to choosing a pump that provides higher or lower flow rate than intended. A pump field test is one way to determine the real/present system curve or pump head (H) against flow rate (Q) (i.e., H-Q curve). A pump field test requires measurement devices to acquire data from the current pumping system. Pressure gauges should be installed in the suction and discharge pipes. Also the elevation of the suction and discharge gauges must be determined with respect to the pump center line. The total pump head can be defined as the "difference between the pressure at the pump discharge (point 1) and that at the pump suction (point 2)" as shown in Fig. 1 [3]. It has three terms: Static Pressure Head, Elevation Difference Head, and Velocity Head or dynamic pressure. Equation (1) gives the total pump head [3].

$$H_{pump\ total} = H_d - H_s = \left(\frac{V_d^2}{2g} + \frac{P_d}{\gamma_d} + Z_d\right) - \left(\frac{V_s^2}{2g} + \frac{P_s}{\gamma_s} + Z_s\right) \tag{1}$$

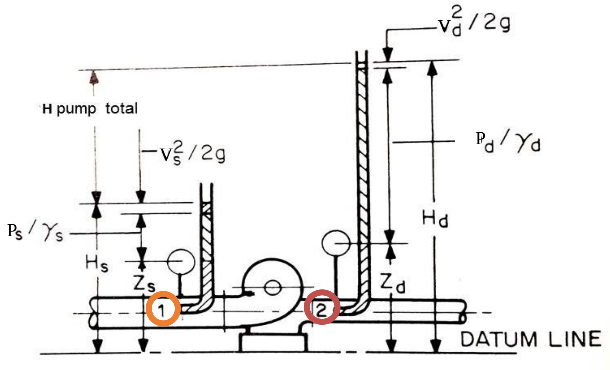


Figure 1: Centrifugal pump total head (reproduced from Ref. 3)

All terms are defined in the Nomenclature. Subscripts d and s shown in Eq. (1) refer to discharge and suction, respectively.

It is well known that the operating point of a pump is the point of intersection of system and pump curves [2]. Therefore, this same point represents the point on the system curve where the pump operation occurs. Once the total pump head is calculated using Eq. (1) and field data is gathered, i.e., system flow rate and pipeline pressure, the pump operation point is determined (see Fig. (2)). The other points can also be located on the (H-Q curve) by understanding the system curve components. One of the components is static head. This head does not change with change in flow rate.

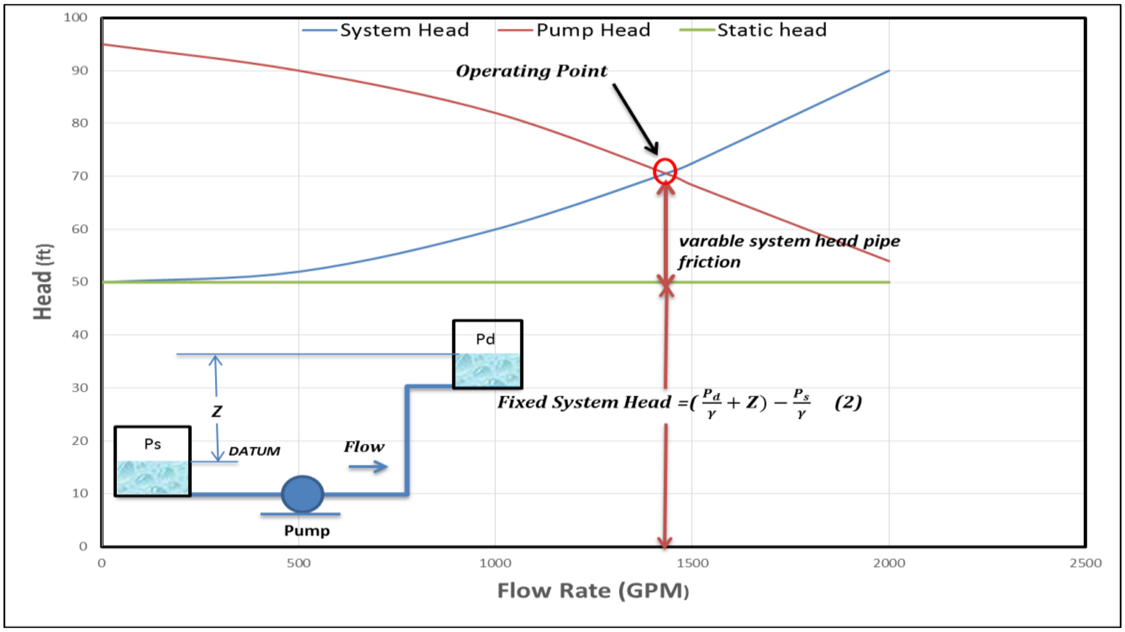


Figure 2: Construction of system total head, or H-Q, curve (reproduced from Ref. 3)

This is why it is also called fixed head. The static or fixed component is equal to the difference between the height of the liquid level in the intake pipe and the height of the liquid level in the discharge pipe, i.e., Z. The difference between the gauge pressure head at the discharge pipe and the gauge pressure head at the suction pipe is then added to Z [3]. Equation (2) [3] in Fig. 2 shows the total static head or fixed system head.

The second component is "friction head". This is related to the fluid velocity, and it increases by increasing the flow rate and vice versa. The "friction head" in a system is related to the velocity head. If the fluid velocity (or flow rate) is doubled, the friction head will increase four times. This can be explained by considering a simple pumping system in which there is no control valve; and the pipe diameter is the same along the pipe length. The Darcy–Weisbach equation [with some modifications to directly relate the friction head to system flow rate] gives the friction head. It is a most useful equation to calculate the pipe friction [3].

$$h_f = f\frac{L}{D}\frac{Q^2}{2gA^2} \quad (3)$$

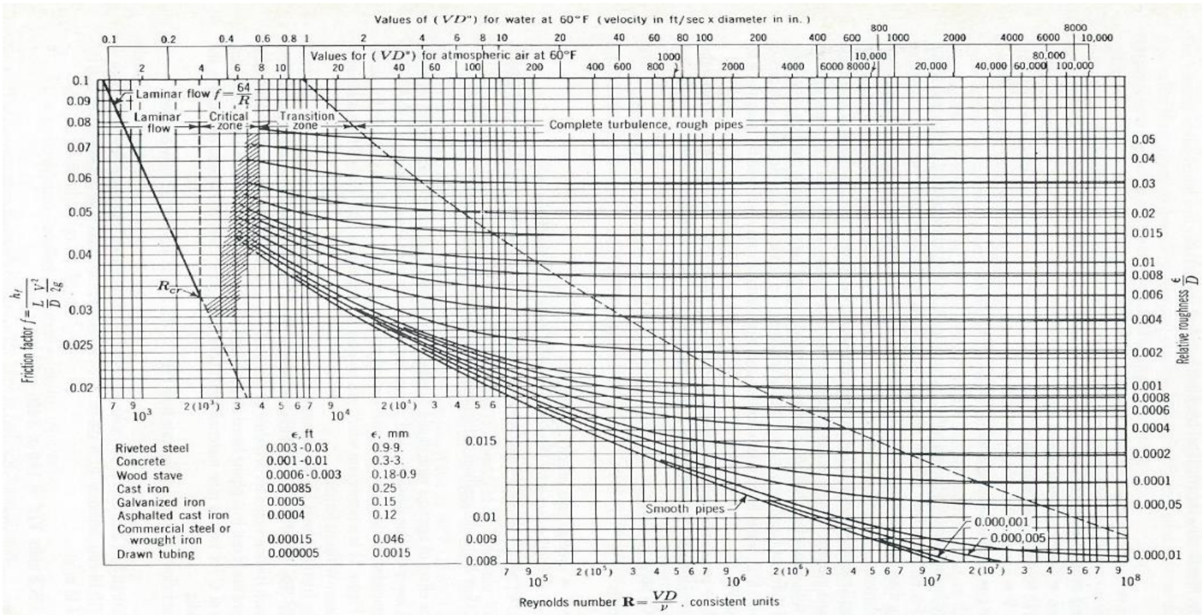


Figure 3: Moody diagram (reproduced from Ref. 3)

It is very important to know that the friction factor f shown in Eq. (3) is not constant, but it depends on Reynolds number (Re) and pipe roughness. However, from the Moody diagram (see Fig. 3) [3], it can be assumed that the friction factor remains constant in the region where flow rate is high, corresponding to a high Re, along with a high relative pipe roughness (see Fig. 3).

Implementing the discussed assumptions, it can be seen from Eq. (3) that the system friction head (h_f) is related to the system flow rate (Q) squared, because all of the other parameters are assumed to be constant, i.e., the friction factor is considered constant, and it does not depend on the system flow rate. The discussed method shows how the system curve changes with the flow rate. Thus, by subtracting the system static head (calculated from the discussion surrounding Fig. 2) from the total pump head measured at the operating point, will give the friction head at that flow rate [2]. Again, it should be noted that the static head of the system curve remains constant even though the flow rate changes. Using the square rule relationship between flow rate (Q) and frictional losses (h_f), or the frictional component in system curve, the other points of the system curve can be located. For instance, in Fig. 2, the static head at zero flow rate is 50 ft, and from the total pump head calculated at 1450 GPM, the total head is 71 ft. Therefore, the total friction head at that flow rate is 21 ft. That means that, at a flow rate of 2 x 1450 GPM, the friction head increases to be 4 x 21 ft. Repeating this process, the system (H-Q) curve can be plotted for a pumping system. The more accurate the measuring devices (i.e., flow rate meters and pressure gauges), the more accurately the system curve can be plotted [2]. Refer to Fig. 4 for an example of system curve.

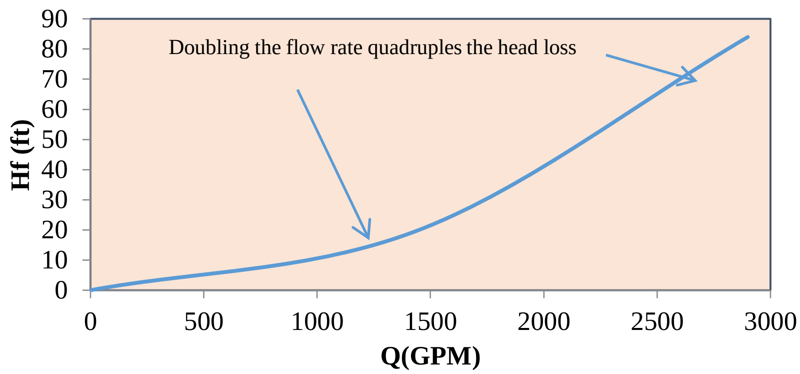


Figure 4: Example friction system H-Q curve (reproduced from Ref. 2)

Once the system characteristic curve is found, options to save energy can be explored. These included either pump impeller diameter trim (the modification of a pump impeller so as to either increase or decrease its diameter), or by using a different pump control method, such as variable speed drives, to perform the job [2]. All of this information helps to choose the right pump. Even though these factors are important, the operating conditions are a top priority for selecting the right pump. Thus maximum accuracy is required so that a maximum possible efficiency can be achieved.

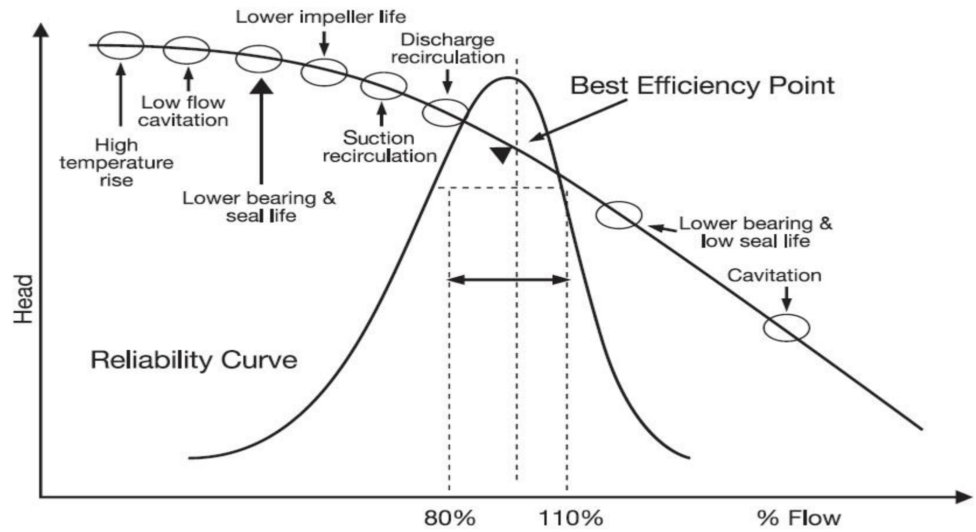


Figure 5: Adverse effects of operating away from the BEP (reproduced from Ref. 4)

For some processes, a pump may be selected to operate over a wide capacity range. That range could be greater than the recommended 80%-110% of the flow at the Best Efficiency Point (BEP). Operating outside this range makes a pump run at a low efficiency which makes the same pump losses reliability (see Fig. 5) [4].

In addition, some pump users may not have their pumps operate at the maximum efficiency point, which commonly happens. However, it is not desirable to have a great difference between the actual efficiency at which the pump is running and the maximum efficiency which the pump can reach (BEP) because of all of the issues shown in Fig.5.

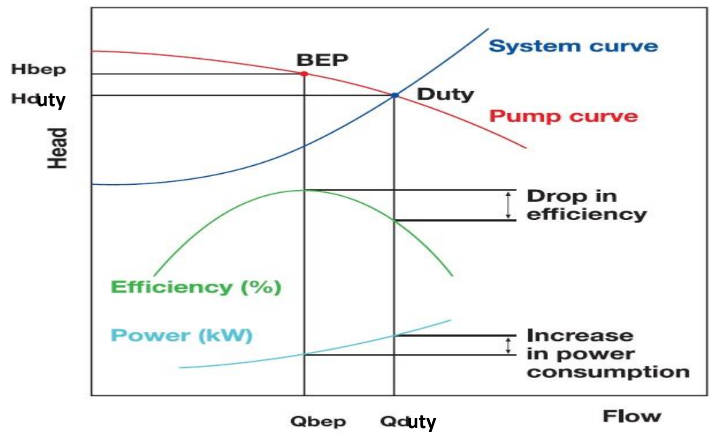


Figure 6: Efficiency drop versus capacity (reproduced from Ref. 4)

Thus, it is very important that pumps run in the range that is 80-110% of the BEP which is called the rated region [4]; or within 70-120 % of the BEP which is called the preferred operating region [5]. Ignoring these guidelines can result in noticeable energy losses, because the energy consumed by a pump can represent from 30% to 90% of the life cycle cost (LCC) of a pump [4] (see Figs. 6 and 7 for more information).

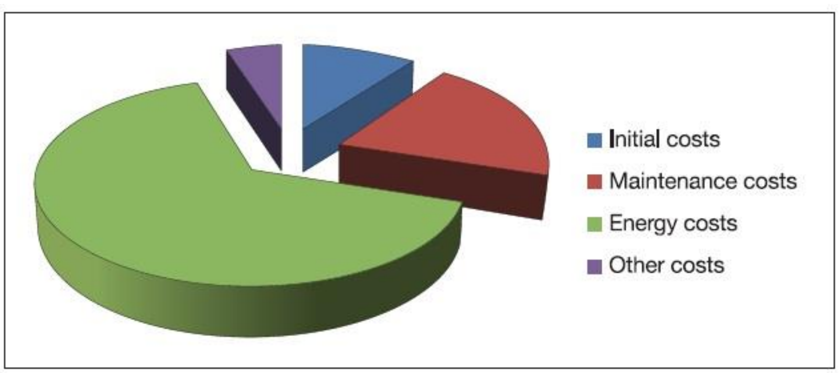


Figure 7: Typical breakdown of pump costs (reproduced from Ref. 4)

The initial cost associated with buying a pump is less important than the operating energy consumed by a pump (Fig. 7), based on LCC analysis (to be explained in detail in later sections of this document). Thus, LCC analysis is a powerful tool which can be utilized to select types of pumps [4].

1.1.1 Constant Speed Pump

A constant speed pump attempts to provide the same flow rate regardless of system demand. “Pump energy consumption is often one of the larger cost elements and may dominate pump Life Cycle Cost, especially if pumps run more than 2000 hours per year” [6]. Moreover, the electric motors of pumps represent 20% of the total energy used in the world [7]. For most industrial systems, between 25% and 50% of the total electrical energy usage is due to pump electrical motors [7].

There is a critical need for selecting and designing the right pump that fulfills the assigned task and the proper control system that contributes to reduced energy consumption. As stated above, constant speed pumps deliver a relatively constant flow rate to the process. However, many HVAC systems do not have a constant load throughout the year because of ambient conditions, occupancy, or the process demand. As a result, control devices must be added to a constant speed centrifugal pump such as a control valve [7]. Obviously, a control valve wastes some energy by blocking the incoming liquid flow from the pump or by returning the excess flow to the pump inlet, which drops the pressure rise in the pumping system. Returning the excess flow back to the pump inlet increases the power consumption of that pump, and blocking the flow causes the system curve to become steeper. The pressure drop associated with a control valve will shift the operating point required for the process to a lower efficiency region of the H-Q curve. Thus, pump efficiency becomes poor. Figure 8 shows how the throttling operating point (1), having a relatively lower efficiency 72%, is shifted from point 2, which has an efficiency of 81%, where the system operates without throttling [6]. Thus control valves, sometimes called “throttling valves” in a pumping system, work inefficiently. However, it is the most common device that is used with constant speed pumps in order to regulate constant speed pumps’ flow [8].

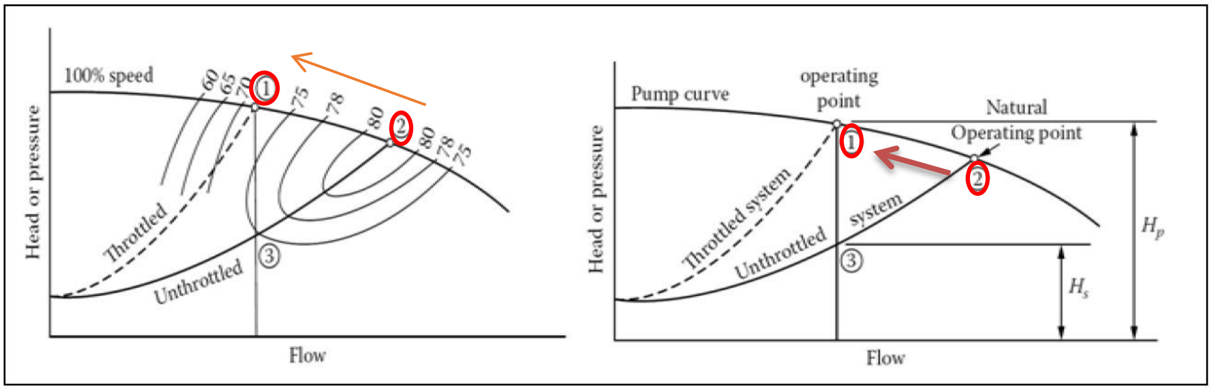


Figure 8: Control valve throttling and pump efficiency. Efficiency contours are shown in left figure. (reproduced from Ref. 6)

Pelikan pointed out [9] that when using a control valve in a large conventional system, there are normally some energy savings due to the the fact that the control valve will shift the operating point on the H-Q curve such that the energy required to run the pump is lower. However, these savings in power are not typically comparable to the savings that are gained by using a variable frequency drive motor to change the pump’s rotational speed.

Additional equipment has to be installed in order to work with a control valve, so that it can respond to the system requirements. A level sensor is a common device that is used with a control valve in order to indicate the amount of flow required by a process. A level sensor can be a standing pipe that is normally installed next to the main tank (i.e., the tank to which the pump provides liquid). In this project, the main tank is the deaerator tank in the KU steam power plant [10]. The level sensor’s fluid shows the same height as the liquid in the deaerator tank. In this standing pipe, there is a float level device which controls an air compressor. The float device will either cause the compressor to increase or decrease the air pressure in order to either open or close the control valve (see Fig. 9). The compresser's piping is connected to a control valve. The higher the level of the float in the standing pipe, the more air will be supplied to the control valve, so that the control valve will close more. The reverse occurs for lower float level and lower pressure. It can be concluded that the compressor has to be "on" all of the time in order to regulate the flow rate by opening or closing the control valve. Thus, more energy will be expended while the compressor is "on", adding to the cost of piping and fittings of the pneumatic air system that is required. Therefore, additional expense will result from using a constant speed centrifugal pump having a control valve [10].

Figure 9: Standing pipe, visual level indicators and level transmitter

Another way to control the flow rate for the process is a by-pass line [11]. By-pass lines can manage the flow rate accurately; and using this control option can avoid the hazard of deadheading. Deadheading occurs when the flow from a pump is completely stopped by some means, such as closing a valve downstream of the pump. On the other hand, this control method has the least energy savings [11]. Figure 10 shows the lost power from using a by-pass option to control the flow rate for a constant speed pump.

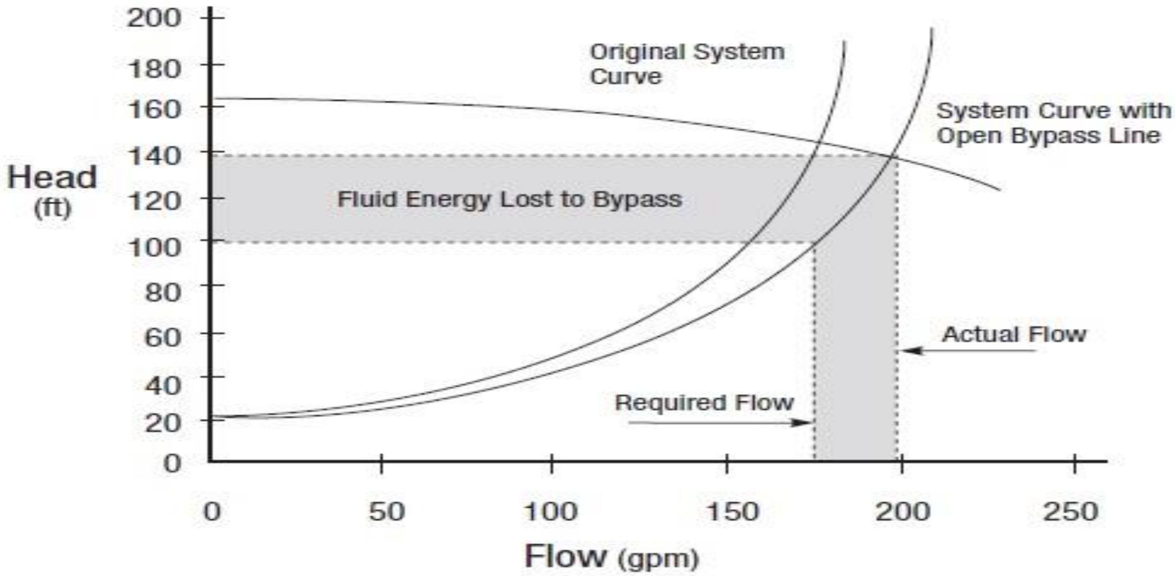


Figure 10: Power lost through a bypass line (reproduced directly from Ref. 11)

The third option that can be utilized while working with a constant speed pump is a hydraulic accumulator [12]. A hydraulic accumulator can save energy because it can store the excess hydraulic energy when system demand is below the design point, and immediately releases this excess energy when needed (i.e., when the system demand is high). According to Mordas [13], a hydraulic accumulator can significantly reduce a pump's size, thus reducing the energy consumed by the pump. As a result, adding an accumulator to a pumping system [according to Mordas] can frequently cut the power consumption from 20% to 70% [13]. However, there are many disadvantages of having an accumulator in a system. Accumulators are expensive and require a space where it can be installed. Besides that, their capacities are limited, and their produced flow is not smooth but is unstable. The flow instability from an accumulator can be explained from the fact that accumulators are used in a hydraulic system that is not run continuously, but "run on an intermittent duty cycle" [13]. In such hydraulic systems, an accumulator is used to store hydraulic energy. That is, any excess flow above the system requirement is kept under pressure, which is then released to compensate for a drop in pressure whenever there is a pressure drop across the pump. This means that the flow rate in a hydraulic system in which an accumulator is present is not constant, but varies often [13].

It has been shown that the costs and losses due to constant speed centrifugal pumps and associated control methods are relatively high [10]. Thus, an alternative solution is needed so that the pumping system can run effectively while reducing the losses and the costs of the system as much as possible. This approach will be presented in this work by using and comparing two different kinds of pumps: a pair of Grundfos variable speed pumps and a Worthington constant speed pump, in the steam power plant of the University of Kansas in Lawrence, KS.

1.1.2 Variable Speed Pump

Variables Speed Drives (VSDs), sometimes called Variable Frequency Drives (VFDs), are not the right solution for every pump application; but they can be the best choice whenever the system demand flow (or system head) varies often. As a result, replacing existing throttling valves with a speed control device can save 5%-50% of system energy usage [6, 14]. VFDs have a good efficiency over a range of flow rates. Even though initial capital cost (purchase price) may be high, it may be justified by offering a substantial savings potential [15]. Constant speed pumps do not work the same as variable speed pumps because the constant speed pumps operate at a flow rate for which the pumps are assumed to be working at their best efficiency [4]. For these reasons, variable speed pumps can be more efficient.

Figure 11 shows an illustration of a VFD [16]. Typically a VFD consists of a rectifier that is used for converting alternative current (AC) to direct current (DC). Then that DC current is fed to capacitors in the VFD in order to smooth the converted current before it enters the next step, i.e, an inverter. The inverter converts the smooth DC into a coarse form of alternating current (AC) [16] that has a variable voltage and frequency using pulse width modulation techniques [17]. Then it is fed to an electrical motor. “The output voltage is controlled so that the ratio between voltage and frequency remains constant to avoid over-fluxing the motor” [17]. Figure 11 shows only a single phase of a three phase current system. Having the speed of the centrifugal pump’s electrical motor changed will vary both the head and pump flow rate capacities [16].

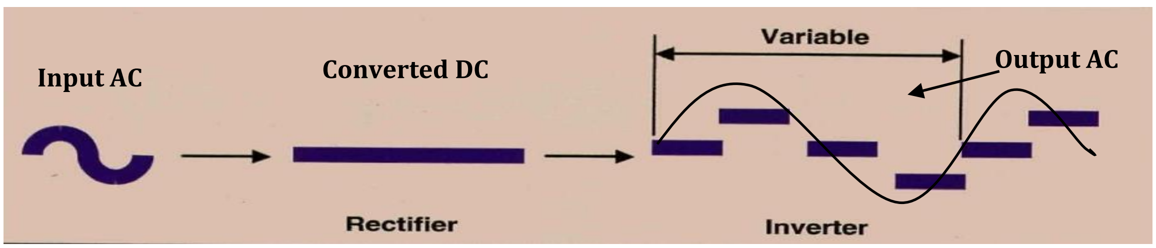


Figure 11: A simplified diagram of how a variable frequency drive works (reproduced from Ref. 16)

The purpose of a variable frequency drive is to change the frequency of the power supplied to the motor. As a result, the speed of the motor also changes. Controlling the pump flow rate and pressure will allow the variable speed pump to match system demand. It is important to have a device that signals the variable speed pump. The most common device used is a pressure-sensing device that is used to maintain the variable speed pump’s discharge pressure at a desired point set by the user. Sometime a flow-measuring device is used to maintain the variable speed pump flow at a relatively constant rate. It is true that a constant speed pump with a control valve can keep the system pressure relatively constant in the discharge pipes according to the system demand - - roughly the same as a variable speed pump operating with a pressure-device controller. However, one must consider that there will be a significant reduction in power consumption when operating a variable speed pump at a low speed [9].

Research has been conducted in Italy to find the best pumping station to fulfil the demands of two irrigation districts [18]. That research demonstrated the ability of variable speed pumps to save energy instead of the constant speed pumps that were already installed in the irrigation fields. Because of the requirement to meet the maximum irrigation demand, the original designers based their designs on meeting the peak irrigation demand. Therefore the irrigation system which includes both the irrigation network and pumping station had been sized to supply the required flow rate at the critical demand point (i.e., the peak demand). However, irrigation districts' demands varied with season; and the peak demand was limited to certain days. Thus during the off-peak time, the constant speed pumping station provided a high-pressure head more than needed for the irrigation networks; and accordingly, the pumps worked inefficiently, i.e., oversized during the low demand time. As a result, research was performed to find appropriate alternatives to save the energy being consumed because of the oversizing of the constant speed pumps. VFD technology was installed in the pumping station. The results of this research showed that the variable speed pumps were able to save energy up to 27% in one of the irrigation districts and saved 35% in the another irrigation district in comparison with the constant speed pump station [18]. As clearly shown by this study, variable speed pumps were able to save a noticeable amount of energy as compared to that of the constant speed pumps. The constant speed pumps were running at constant speed regardless of the system demand.

In Queens, New York, another example of using VFD pumps in the Astoria Power Generating Station showed not only reduced energy consumption, but also a decrease in the annual consumption of cooling water from the river by the plant [19] . Changing constant speed pumps in three units of the water recirculation part of the plant to variable frequency drive pumps was mandatory due to New York Department of Environmental Conservation (DEC) regulations. The reduction in water flow through the Astoria power plant helped to reduce “impingement and entrainment (IM& E) of aquatic organisms and minimize environmental impacts” [19]. However, reducing circulating cooling water flow though a plant can negatively affect the plant’s thermal efficiency. The purpose of the variable speed pumps was to reduce the total discharge cooling water flow rate; but the reduction of the cooling water had to be adjusted within certain limits in order not to have the power plant work at poor efficiency, i.e., working “on acceptable performance penalties” [19] and to follow the regulations of the DEC which mandated that the power plant meet targets of maximum temperature rise between the inlet and discharge cooling water. The goal of using variable speed pumps was met, and there was a significant reduction in environmental impact. This achievement could not be achieved by using constant speed pumps [19].

In Sweden, Alfredsson and Bokander [15] showed that, in some applications, variable speed pumps could save € 78,681 over a lifetime of ten years. In their study, they stated that variable speed pumps work perfectly for a system head that varies often and has a relatively low static head. In that study, a simple pumping system was considered in which a heat exchanger was used to heat up the working fluid for process requirements, and a control valve was used with a constant pump.

The static head for the system was 10 m with a maximum flow rate 850 m^3/hr. In addition, a 10% factor of safety was added to cover any future project expansion. As a result, the design flow rate was selected by rounding up the numbers to be 1000 m^3/hr. The total head losses (static head, friction losses in pipes and fittings, friction losses in the heat exchanger, and control valve head loss) for a flow rate of 850 m^3/hr was found to be 40 m. The total head losses included a 7 m factor of design safety and another 5 m drop in pressure head across the control valve. Three operating points were selected for the constant speed pumps (C1, C2-a, C3-b); where C1 was the operating point for a constant speed pump with an impeller diameter of 355 mm. The operating point of this constant speed pump was a flow rate of 850 m^3/hr and a discharge head of 33 m (i.e., without any factor of safety added at this operating point). C2-a was an operating point for a second alternative constant speed pump with an impeller diameter of 385 mm. The operating point of this constant speed pump was a flow rate of 850 m^3/h and a discharge head of 43 m (i.e., including a factor of safety). C2-b was an operating point for a third alternative constant speed pump with an impeller diameter of 385 mm. The operating point for this constant speed pump was a flow rate of 1000 m^3/hr and a discharge head of 40 m (i.e., maximum possible flow rate and a factor of safety).

On the other hand, two operating points were selected for two alternatives using variable speed pumps (V1, V2), where V1 was an operating point for a variable speed pump alternative that operated at a maximum flow rate, but no factor of safety was added (850 m^3/hr and 28 m head). The discharge pressure head for this alternative was only 28 m because of the fact that the control valve was not used. Therefore, the local pressure head of the control valve was removed, the total head loss was reduced to 40-12= 28 m of head (for the case of not having the control valve in the system); and V2 was an operating point for anther variable speed pump alternative. The operating point of this variable speed pump was at the maximum possible flow rate and discharge head (including a factor of safety), 1000 m^3/h and 35 m, respectively.

Life Cycle Analysis (LCC) (to be explained in Section 1.2) was performed to compare the pumps' cost-effectiveness. In order to calculate the overall LCC results, assumptions were made. The interest rate was assumed to be 6%, the inflation rate was 3%, and the total calculation time was 10 years. (For other input LCC calculation data input, see Ref. 15). After calculating the LCC for all alternatives of the constant speed pumps and variable speed pumps, the study showed that the LCC costs for the constant speed pump option were: C1: €243134, C2-a: €302256, and C2-b: €320678; while, for variable speed pump options, costs were: V1: €223575, and V2: €305617. As stated previously, this study showed some significant savings in the case of the variable speed pump. So as the results demonstrated, the LCC benefit when comparing options C2-a and V1, was € 302,256 – € 223,575= € 78,681 or 26% of constant speed pump costs. For Ref. 15, the variable speed pump was the best option and resulted in a lower LCC as compared with that of the constant speed pump for the same case.

Alfredsson and Bokander pointed out that, even though there are some energy savings in each case, variable speed pumps have to be carefully studied before deciding to purchase them, because they are not always a lucrative solution. One such case is when the static head of a system curve is more than one half of the total losses head (i.e., $H_{static} > \frac{1}{2} H_{system}$) [15].

Another study [20] focused on using variable speed pumps as a primary only (p-only) pumping system for a chilled water plant. For many years, the chilled-water system was generated based on constant-flow-primary, i.e., using constant speed pumps on the chiller side, and variable-flow secondary (p-s) system, i.e., using variable speed pumps on the load side of the chiller plant (the last is a typical design in an HVAC system). This study was made to compare the energy consumption of the three pump operating configurations (i.e., primary-only, primary-secondary, and variable-primary pumping systems). In the first pumping arrangement, as shown in Fig. 12, the single primary set of constant speed pumps was responsible for both the plant and the system pressure drop (i.e., load). Therefore, the constant speed pumps was responsible for providing return chilled water from the load to the two chillers shown in Fig. 12 and also responsible for providing the chilled water to the load.

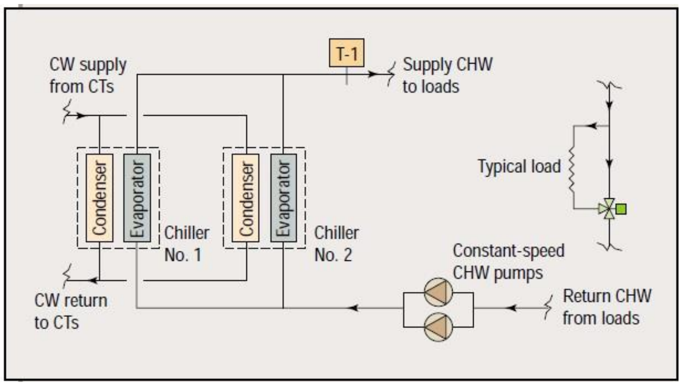


Figure 12: Constant-flow, primary-only system configuration (reproduced from Ref. 20)

For the second pumping arrangement (constant flow primary- variable flow secondary) shown in Fig. 13, the constant speed pumps supplied chilled water to two chillers at a relatively constant flow rate. However, the secondary side [distribution] flow varied according to the load.

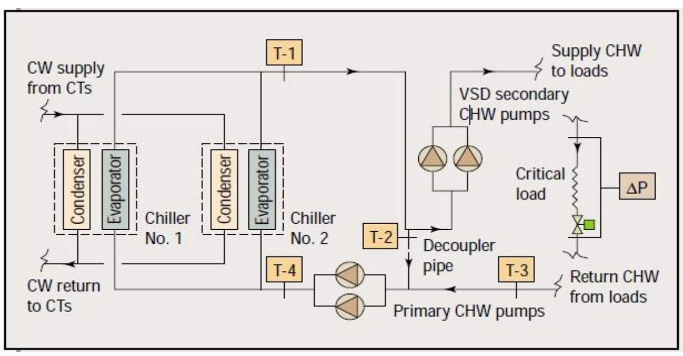


Figure 13: Constant-flow-primary/variable-flow-secondary system configuration (reproduced from Ref. 20)

In this pump arrangement, potential energy savings can be achieved by varying the flow on the secondary side [distribution] of the system corresponding to the load. This option was not available for the first pump arrangement (Fig. 12). The third pumping arrangement (variable-flow primary-only) is shown in Fig. 14. This pump configuration was similar to the constant–flow primary-only arrangement (Fig. 12). However, the variable speed pumps were employed for both chillers [plant side] and the load [distribution side]. The study simulated the work of variable-primary-flow system performance in order to compare it with the first and second pump arrangement performance in operation. Again, the study was seeking the most economical and energy saving pump arrangement as well as to change the standard method by which a chilled-water-plant could operate.

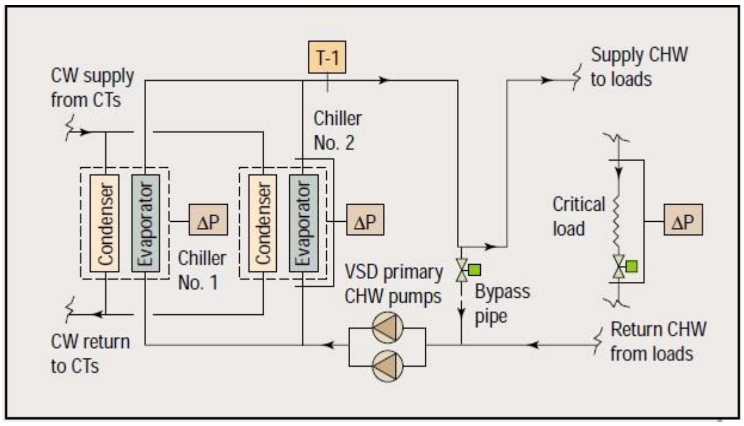


Figure 14: Variable-flow, primary-only system configuration (reproduced from Ref. 20)

The input data for the simulations was taken from a real building which had five-stories with an unconditioned basement. The working area was roughly 28000 ft^2 per floor. The main effect of having a variable speed pump in the primary chilled water plant circuit was that it permitted the primary chilled water plant to work at a reduced flow rate when the cooling load was reduced. Thus energy could be saved in the primary pumping circuit. After taking data for one season (cooling season ran from April 1 to October 31), it was shown that the constant-primary-flow-system used the highest energy, while constant-flow-primary/variable-flow-secondary system used less energy than the constant-primary-flow-system; and finally, the variable-primary-flow-system used the least energy. To illustrate the energy consumption for chilled water pumping as a fraction of total energy consumption by the plant, the following results were computed: for one-chiller, the constant-primary-flow operating case consumed approximately 20 % of the total energy for chilled water circulation, whereas 10 % of the total energy was consumed by the constant-flow-primary/variable–flow-secondary case. Finally, only 5 % of the total energy was consumed by variable-primary-flow system [20]. From this study, variable speed pumps had effectively contributed in energy savings as compared with the other available options. This is an indication of how variable speed pumps can play a significant role in energy savings for such applications.

Despite the previously discussed studies which have shown great benefits from using variable speed pumps in pumping systems, some researchers have found that there are no advantages in using variable speed pumps, or that, in fact, variable speed pumps consume more energy than constant speed pumps if the two different types of pumping systems are sized correctly for a specific application [21].

Austin [21] compared a variable speed pump with a constant speed pump. For this comparison, the actual pump characteristic curves for both the variable speed pump and the constant speed pump were used, with a system demand of 1200 GPM for 12 hours. After that 12 hours, the demand decreased to 100 GPM for the other 12 hours per day. Using the constant speed pump to supply the 1200 GPM with a total head (static head plus dynamic head) of 231 ft, this pump consumed 100 HP. However, when running the variable speed pump for the same demand and same total head, the variable speed pump consumed 103 HP. The energy consumed was without considering the drive loss and the motor loss, because the motor will run on a pulsing DC voltage which yields a lower motor efficiency. The extra power used because of the additional losses was in the range (3% to 5%).

For the 12 hours when the demand was lowered to 100 GPM and a total head of 231 feet, the constant speed pump with a control valve consumed 42 HP. On the other hand, the variable speed pump consumed 38 HP at a reduced speed of 3280 RPM. By considering the additional motor losses mentioned above, the total energy consumed by the variable speed pump became 39.14 HP. Therefore, the variable speed pump consumed 103 HP when it was meeting the system demand of 1200 GPM. That is equal to 12 gallons per HP. So when the variable speed pump met the second shift demand of 100 GPM, it consumed 38 HP. That is equal to 2.63 gallons per HP. Austin [21] concluded that "at 100 GPM using 38 HP, variable speed drive is burning 456% more energy per gallon than when the pump is running at constant speed at 1200 GPM" [21]. His conclusion can be clarified by examining the two operating shifts of the variable speed pump. In the first operating shift when the demand was high, the variable speed pump consumed only 1 HP for moving 12 GPM, whereas it consumed 1 HP for moving 2.63 GPM when the demand was low.

According to Austin [21], the control valve will waste some energy when the shifted operating point is compared with the best operating point (the point where the pump works at its best efficiency). However, when the amount of lost energy from using a control valve was compared with amount of energy lost when using a variable speed pump, the difference was low, just 1 HP. Austin explained that, when the pump flow was restricted by a control valve, the restriction did not make the motor work harder as many people think. Instead by looking closer at the pump curve, restriction of the pump flow reduced the power consumed by the motor [21]. However, the pump efficiency decreased when the flow was restricted (see Fig. 8 as an example) [6], Also, this study did not consider that the flow restriction causes the pipeline upstream of the control valve to be over pressurized. That can affect the pipeline and the pump itself because the pump will be back pressurized which can directly affect the mechanical seals of that pump unless there is a recirculation line that recirculates the excess flow back to the inlet side of that pump [22]. This type of situation was witnessed during data gathering for this KU steam power plant project when the vent condenser was shut-off, and the Worthington constant speed pump was working during the low demand time, and unintentionally the recirculation line was closed. The pipeline upstream of the control valve became pressurized and that caused leakage from spots in which the pipes had some old damage.

Austin also indicated that variable speed pumps do not just consume more energy, but they also have disadvantages that can cause even more side effects for the pumping equipment. Side effects such as: “Pulsing DC voltage, EDM currents, critical speed vibration, harmonics, Radio Frequency Interference” [21] can cause pumping equipment damage that needs to have immediate technical service. Finally, if no energy can be saved by using variable speed pumps, one can easily choose a constant speed centrifugal pump which is less expensive and has less complicated flow controls (for instance, a control valve) and then save money without being concerned with the side effects that involve using variable speed pumps, Austin affirmed [21].

Finally, there are other factors that can cause a variable speed pump not to be desirable for specific applications. Such applications are when the system needs, for most of the time, a constant pressure and flow rate. One should not select a variable speed pump when pressure is a substantial factor and needs to remain constant most of the time. That can be explained by examining the second affinity law, Eq. (4), which shows that the pump head “Pressure head” is proportional to the square of the motor speed. Thus, whenever the speed decreases to half of full speed, the pump head will reduce to one-fourth of full-speed pressure. However, when the system demand changes from a low-head, low flow situation to a high-head, high-flow situation, then one should think of using a variable speed pump to meet such demands [23]. The goal of this project is to investigate the conditions under which the variable speed pump is applicable, and where there might be potential power savings in a steam power plant.

$$\frac{H_2}{H_1} = \frac{n_2^2}{n_1^2} \quad (4)$$

All terms used in Eq. (4) are defined in the Nomenclature; and subscripts 1 and 2 denote two different pump speeds.

1.2 Life Cycle Cost Analysis

Using life cycle cost analysis (LCC) can help companies enhance energy efficiency and reduce wasted energy, not only for pumping system applications but for many other applications, by identifying the most cost-effective alternatives from different options when installing new equipment [24]. LCC analysis can be defined as "the total cost of ownership of machinery and equipment, including its cost of acquisition, operation, maintenance, conversion, and/or decommission" [25].

When considering a new project, procurements costs, i.e., equipment costs, have been considered as the main focus, or sometimes the only factor taken into consideration, in order to choose equipment or a system [25]. However, it appears that this is not always the right decision from a financial point-of-view. We should consider the cost over time and procurement strategies. Judging based on the lowest initial costs, does not consider maintenance and other problems in the long term [26]. For this reason, using LCC analysis helps to identify whether or not operational savings are adequate to justify the investment cost. Thus, LCC analysis becomes very significant when used as a management tool to compare among multiple alternatives, as it will help to show the most cost-effective solution [24]. However, according to Ashworth [27], LCC analysis may not always provide reliable predictions. He reasoned that "There is little evidence to support the view that previous life cycle costs have produced reliable forecasts" [27]. Thus, estimated values far in the future may result in inaccurate predictions [27]. Because of the difficulty of getting the exact data for each component of LCC, Barringer and Weber pointed out that "LCC is not an exact science, everyone gets different answers and the answers are neither wrong nor right" [28]; and the results of calculating LCC are not precise.

According to 1994 statistics in the U.S. industrial sector, over 679 billion kWh of electricity per year were consumed by electrical motors, and 25% of that total consumed energy was by pumping systems [14]. Because of that, it becomes very important to rely on LCC analysis so that the most cost-effective pumps are considered in an application. However, it is very common to focus on the initial cost of a pump when considering a new project. This is not a complete picture, as the initial cost is typically a small portion of the total operational costs over its life. The total cost starts at the purchase of the equipment and goes through disposal cost of that equipment. So, the total LCC depends on the type of application and length of time used (15 to 20 years for pumping systems) [14]. Therefore, in order to minimize the total LCC, it is crucial to consider other costs, such as energy to generate and maintain, which can overshadow the initial cost [24]. With all other costs taken into account, the LCC equation for a pumping system can be written to represent each cost element as follows [24]:

$$\boldsymbol{LCC = C_{ic} + C_{in} + C_{e} + C_{o} + C_{m} + C_{s} + C_{d}} \quad (\mathbf{5})$$

Equation (5) represents all elements of the LCC with all terms defined in the Nomenclature.

LCC analysis can be performed by using two methods: manually building worksheets or using existing computer programs. In the United States, with Federal Energy Management Program (FEMP) sponsorship, the National Institute of Standard and Technology (NIST) has devised four programs that can greatly help in providing "economic analysis of proposed investments in buildings and building systems which are intended to reduce long-term operating costs: BLCC, QI, DISCOUNT, and ERATES" [29]. These programs are valuable for identifying the most cost-effective alternatives. The alternative that has the lowest LCC result can be chosen from multiple possible projects or designs in order to save/conserve energy and water. These programs can be used by federal, state, and local governments, and by the private sector. At the beginning of each federal fiscal year, these programs are automatically updated so that current estimated future values of the FEMP rate of interest and the energy price escalation rate can be used in calculating LCC appropriately [29]. For this project, the BLCC program will be employed in order to build the LCC analysis.

For a specific analysis, not all of the costs in Eq. (5) have to be present. That can depend on the type of application and the user's decision. However, in order to make the LCC analysis, each cost element should be considered. After all costs in Eq. (5) have been determined for all available pumping system alternatives or designs, one can use the U.S. FEMP program (BLCC) to evaluate the LCC for each alternative; or one can use any program with which the user is more comfortable. Once the LCC is found, a plant manager can make the final decision in choosing which alternative is the most cost effective, i.e., the one which has the lowest LCC [24].

1.3 Steam Power Plant Description and Scope of Work

This project is directed toward arriving at the most cost effective method that can be used to control the water flow to the deaerator tank sitting in the basement of the KU steam power plant. The comparison will be between a Worthington D-824 constant speed centrifugal pump (see Appendix A1 for more information on this pump) shown in Fig. 15 and pair of "BoosterpaQ® Hydro MPC

CRE 15-3” variable speed pumps provided by Grundfos Pumps Corporation (see Appendix A2 for more information on these pumps) shown in Fig. 16.

Figure 15: Worthington D-824 constant speed centrifugal pump in KU Power Plant

Figure 16: BoosterpaQ® CRE 15-3 variable speed pumps in KU Power Plant

The steam power plant located at the University of Kansas, Lawrence campus was used as a source of electricity in the past. However, with time, the power plant changed to supply the university with steam only. That is why the name used in this document will be “steam power plant” instead of “power plant,” because there is no turbine in the plant generating electricity. The steam generated from the steam power plant is used to provide hot water to the University’s buildings. The other function of the steam power plant is to provide the required steam for HVAC systems all around the university and to release some amount of steam to sound the change of classes.

The University’s campus steam lines supply of steam should be at a constant pressure of 90 PSIG, since some equipment requires that steam pressure. However, the steam power plant has to keep the steam pressure at the boiler pipes constant at 170 to 175 PSIG in order to maintain 90 PSIG in the university supply’s lines. The steam power plant’s production varies from 20,000 – 70,000 lb_m/hr of steam during the year [10]. Steam production is not constant because the load (steam demand) does not remain constant, and depends significantly on two factors: 1) the outside weather “temperature”, and 2) the number of people on campus. Therefore, in order to maintain constant steam pressure in plant system lines and the campus steam lines, the boilers have to generate the demanded flow rate.

In the steam power plant, there are a total of four condensate pumps including the pumps that are being compared in this thesis, four boiler feed pumps, and two booster pumps. The booster pumps are used to increase the deaerated water pressure before it enters the boiler feed pumps (see schematic in Fig. 17).

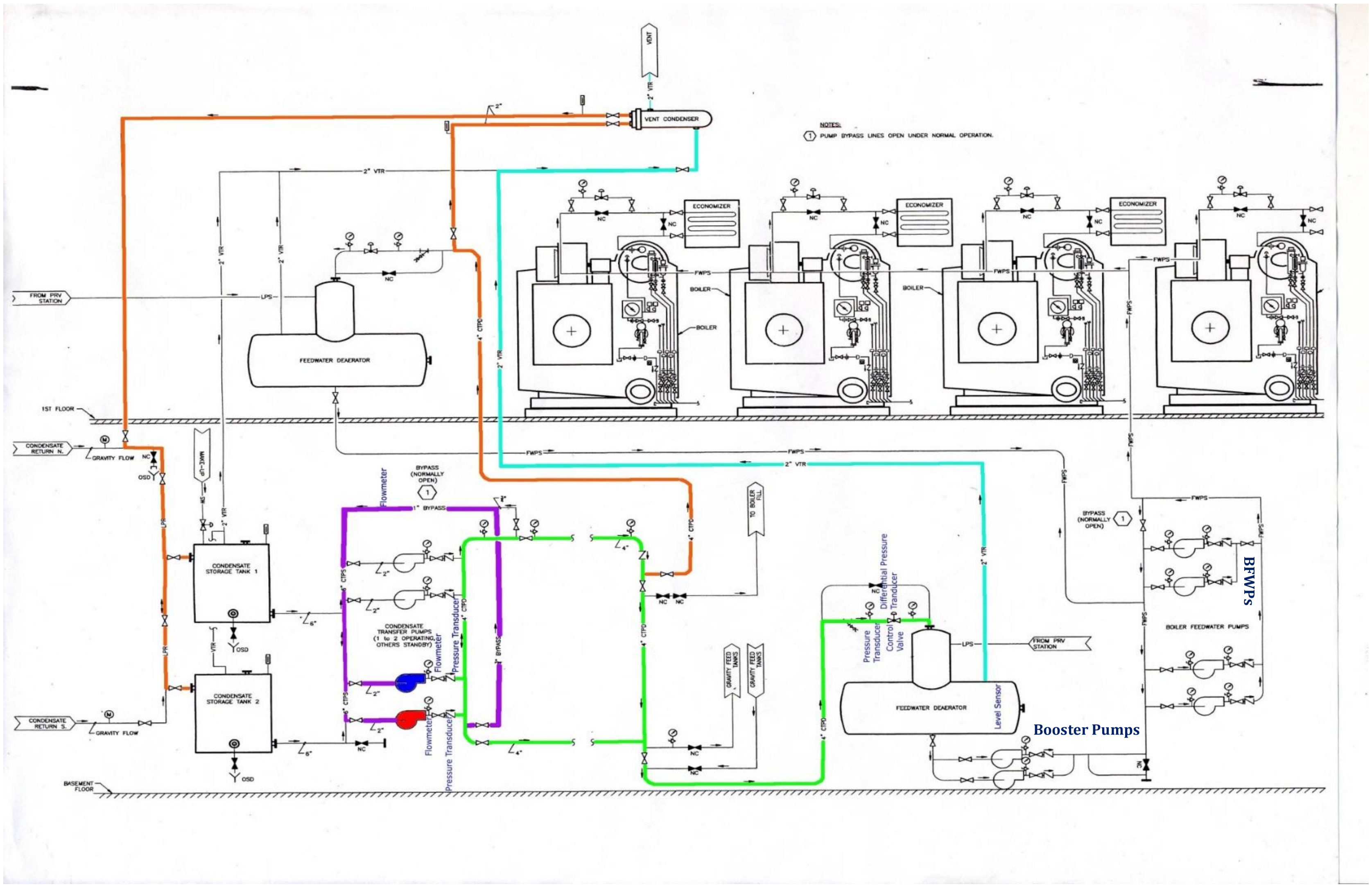


Figure 17: The University of Kansas Power Plant Schematic. (Red Pump= Worthington D-824; Blue Pump= CRE 15-3; Green lines= Deaerator water supply; Light blue line= Steam line to vent condenser; Orange lines= condensate water supplied to vent condenser and back to condensate storage tanks; Purple lines= Bypass to return water to pumps suction side (reproduced from Ref. 10)

The schematic shown in Fig. 17 represents the whole steam power plant, where the red colored pump is the constant speed pump, and the blue pump represents the variable speed pumps (comprised of two pumps working in unison). The green line designates the 4” pipelines that provide condensate water to the deaerator tank in the basement of the power plant. It can be seen that there are two deaerator tanks: one in the basement which is currently operating and one on the first floor, which had not been used for entire time period of this project. This is because the deaerator in the basement fulfills the boilers’ demands, and it has a higher capacity than the one in the first floor. Not all of the water provided by the condensate pumps ends up in the deaerator tank (DA); but some of it moves through a 2” pipe (represented by the orange line) up to a vent condenser where the water is heated up, then sent back to the main two storage tanks (approximately 10000 US gallon total) [22] which supply the condensate pumps.

Some of the steam produced by the boilers is directed to the deaerator tank in the basement (this line is not shown in the schematic) in order to preheat the condensate water before it enters the boilers so that the boiler efficiency increase. Some excess bled steam [and entrained air] from the DA tank is released/vented to the vent condenser by way of the light blue pipeline. As the saturated steam enters the vent condenser, it condenses on the outsides of the tubes present within the vent condenser. The condensate steam is sent back to the main storage tanks through the orange 2” pipe line as shown on the schematic of Fig. 17. The condensate steam is supposed to help the steam power plant save money by not buying that amount of water from the city of Lawrence [10]. However, when the power plant technicians were asked about the amount of water provided by the vent condenser, the reply was “not a significant amount, but there is water provided by the vent condenser” [30]. The amount of water that the vent condenser provides from condensing the excess steam that is escaped from the DA tank is unknown because there was no flow meter installed in the condensed return steam line from the vent condenser. Therefore, this project has no information on that amount of water. From estimation, it can be assumed that this amount of water is roughly 5% of the total vented amount of steam. Some of the steam that does not condense inside the vent condenser, along with the non-condensable gases [coming from the deaerator tank], are released to the atmosphere.

The recirculation line, shown in Fig. 17 as the purple pipeline, returns any excess water to the pumps' suction pipeline. However, little flow has been seen in the recirculation pipeline during the period in which the data was gathered for this project because the steam power plant staff keeps this pipeline closed most of the time, and opens this pipeline valve in two situations: 1) if the vent condenser valve is closed, all of the provided water from the condensate pumps, including the pumps being compared in this project, will be directed to the DA tank. Thus in order not to overfill the DA tank, the control valve will block the excess flow. This situation increases the pressure on the upstream side of the control valve (green line on schematic in Fig. 17). Therefore, the steam power plant staff decides to open the recirculation line. 2) If the steam demand is relatively low, the control valve will try to block the excess flow in order not to overfill the DA tank (same type of situation as in situation 1). Therefore, in order not to over pressurize the discharge line (denoted by the green line in in Fig 17) by this situation, and back pressurize the pump that is on duty, the steam power plant staff also decides to open the recirculation pipeline valve [22] to relieve the excess pressure.

Furthermore, there are pipelines that return the condensate steam from all around the campus to the storage tanks. These pipelines, labeled as "condensate return" in the schematic (Fig. 17), can be seen on the left side of the storage tanks.

For normal steam power plant operation mode, these are all of the related pipelines as shown in the schematic in Fig. 17. However, when the variable speed pump is working in level control mode and when the vent condenser valve is closed (no water is going to the vent condenser), the excess steam from the deaerator tank [that normally goes to the vent condenser] is vented to the atmosphere via pipelines that are not shown in the schematic in Fig. 17.

Finally, a “Heat Recovery system” similar to the one shown in Fig. 18 [31] is installed in the basement of the steam power plant. The heat recovery system reclaims the lost energy from “Boiler Blowdown.” When the condensate water evaporates in the boiler drum, condensate water impurities remain in the bottom of the boiler’s drum. This can cause sludge or sediment accumulation in the boilers. As a result, the overall boiler heat transfer decreases [32]. Therefore, in order to reduce these effects, the Total Dissolved Solids (TDS) level has to be minimized to an acceptable level [3000-4000 µs/cm] [22]. In order to keep that level of TDS in the specified range, the steam power plant staff from time to time discharges water from the boilers.

There are two kinds of boiler blowdown: one is the “surface or skimming blow down” [32] which attempts to eliminate the dissolved solids that accumulate near the liquid surface [32]. This process is continuous during the boiler’s operation. The second is called “mud or bottom blowdown” [32]. This is not a continuous process, and is carried out for a few seconds every several hours. The goal behind this process is to remove suspended solids that settle as sedimentation from the boiler water and develops a heavy sludge [32]. These two kinds of blowdown are used in the KU steam power plant boilers.

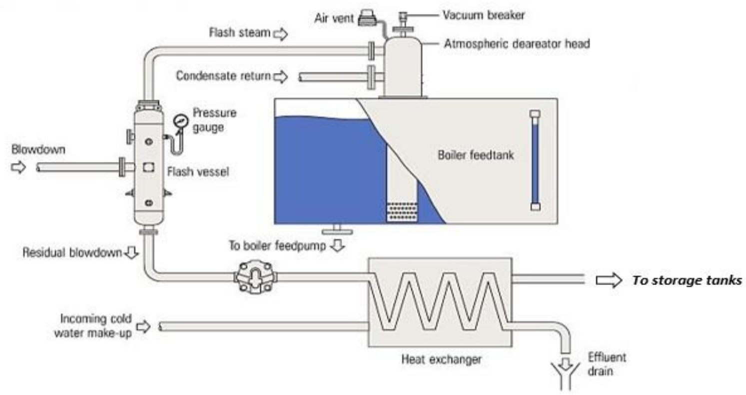


Figure 18: Energy recovery using heat exchanger (reproduced from Ref. 31)

Figure 18 shows how to reclaim the energy lost due to boiler blowdown. In Fig. 18, the flash vessel is used to separate the steam from water; and that steam is directed to the boiler feed tank (deaerator) which is installed next to the flash vessel. The other part of the heat recovery system is the heat exchanger. Instead of wasting useful energy from the residual blowdown (see Fig. 18), a heat exchanger is installed for heating the make-up water which is relatively cold (~ 60 ^{o}F). Then, the heated make-up water is sent to the main storage tanks. The heat exchanger raises the make-up water temperature up to 80 ^{o}F. After that, residual blowdown hot water becomes relatively colder. It is discharged to an underground tank in order for the sludge to be deposited at the bottom of the tank, and the water cools even more. Lastly, the water is moved by gravity to the city sewer. (This system is not shown in the schematic in Fig. 17.)

The four condensate pumps, including the pumps being compared in this project, were installed to provide water to only DA tank #2 in the basement. However, to effectively utilize the excess steam from the DA tank, a vent condenser was installed in 2011 on the top of the first floor, approximately 25 feet from the ground level of the first floor. The DA tank's water is moved to boiler feed pumps by two constant speed booster pumps (shown in Fig. 17 next to the DA tank) at normally 28 to 30 PSIG. The boiler feedwater pumps (BFWP) raise the water pressure to 330 to 350 PSIG; and the compressed water then enters the boiler to generate steam. See Fig. 17 for BFWPs' location.

The deaerator tank [or open feed water heater] heats the water by direct contact with the steam extracted from the boiler. Typically, this would be the steam extracted from the turbine of a power plant. However, because there is no turbine in the steam power plant, the steam is bled from the boiler, yielding a similar concept for the DA tank. Thus, it is known as a direct contact feedwater heater. The DA tank's benefit is to preheat the feedwater and subsequently increase the boiler efficiency. Usually, the DA tank is kept pressurized so that no air leaks into the tank. The water enters at a specified pressure and is heated to a specific temperature in order to decrease the concentration of oxygen as much as possible (see Fig. 19 for more information) [33].

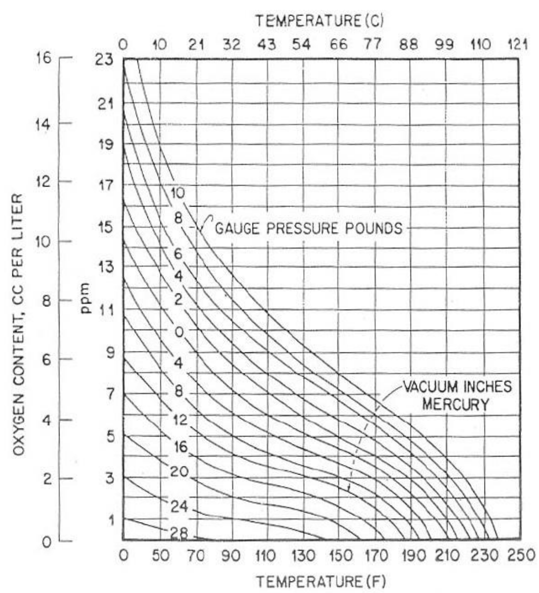


Figure 19: Solubility of dissolved oxygen at various pressure/temperature conditions (reproduced from Ref. 33)

Figure 19 shows the maximum oxygen concentration that can be found in water over a limited range of water temperatures and pressures [33].

DA tanks are mostly designed to keep oxygen concentration in the outlet at less than 0.005 cm^3/L [34], as oxygen has undesirable effects on equipment such as boilers [22]. The steam power plant operators also add chemicals, such as the oxygen scavenger “NAlCO BC1851”, to eliminate as much oxygen as possible from the feedwater in order to avoid corrosion in the boiler tubes (downcomers, risers, drum, and header). Normally, the water in the DA tank is at a saturated state or close to a saturated state. Pumped water at saturated conditions may result in cavitation “flashing on the back side of the pump vanes” [34]. For this reason, the DA tank is located at an elevation above the booster pumps, used to move the water to BFWPs (see Fig. 17), in order to have the necessary pressure at the pumps’ inlets [34]. In addition to the DA functioning as described above, the DA tank is used as a storage device, because it stores water to keep feeding the boilers with hot deaerated water in order to generate the steam at a constant pressure. Most of the DA tanks are designed to store sufficient feedwater so as to have reserve boiler rated capacity feedwater for 10-20 minutes, in case of emergency [22, 34, 35].

There are three types of DA feedwater heaters: 1) spray-type deaerators 2) tray-type deaerators and 3) combination spray-tray deaerators.

The steam power plant in the University of Kansas has the second type of deaerator tank [tray-type deaerator] in the basement, and the first type on the first floor – which is not on duty. In the second type of deaerating feedwater heaters, the feedwater is directed over horizontal perforated trays and descends from tray to tray while encountering the upcoming extracted steam [from the boiler]. As the extracted steam scrubs the feedwater, the non-condensable gases and some of the extracted steam are released to the atmosphere [34]. Figure 20 shows a schematic of a tray-type deaerator tank [10].

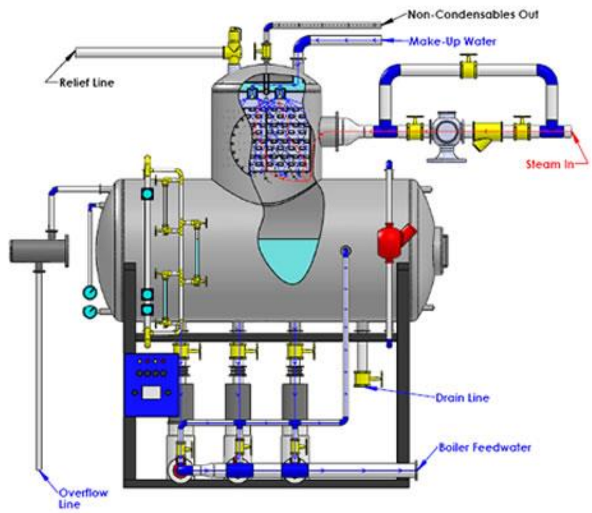


Figure 20: Tray-Type deaerating feedwater heater tank (reproduced from Ref. 10)

In the steam power plant, the escaping steam and non-condensable gases [with their remaining energy] pass through the first floor vent condenser instead of getting released to the atmosphere. The reason for this is that the vent condenser behaves as a heat recovery device, and condenses the escaping steam from the deaerator to replace some of the make-up water that must be purchased from the city of Lawrence. The vent condenser captures much of the steam and non-condensable gases' energy before they are released to the atmosphere by heating up the condensate water, which is provided by the condensate water pumps. The condensate water is heated in the vent condenser and sent back to the main storage tanks. This process can be seen on Fig. 17 by following the orange line. This heated water raises the temperature of the rest of the condensate water in the storage tanks. The overall temperature increase helps to improve the steam power plant's efficiency, i.e. the boiler efficiency improves [10].

The other benefit from the vent condenser is to trap and retrieve any water mixed with the non-condensable gases. The non-condensable gases have steam entrained that escaped from the deaerator tank. Instead of just venting the entrained steam to the atmosphere, the steam rejects heat to the relatively cooler condensate water supplied by the condensate pumps [as discussed above]. Then the steam condenses on the outside surface of the vent condenser tubes in which the relatively cold condensate water flows within the vent condenser shell. Eventually, the condensed steam is sent back to the main storage tanks by gravity (see Fig. 17 and Fig. 21). Therefore, in order to compare the variable speed pumps and the constant speed pump, two case studies were analyzed. However, the first case was excluded from LCCA due to the variable speed pump performing more work than the constant speed pump. This situation will be explained later in this document.

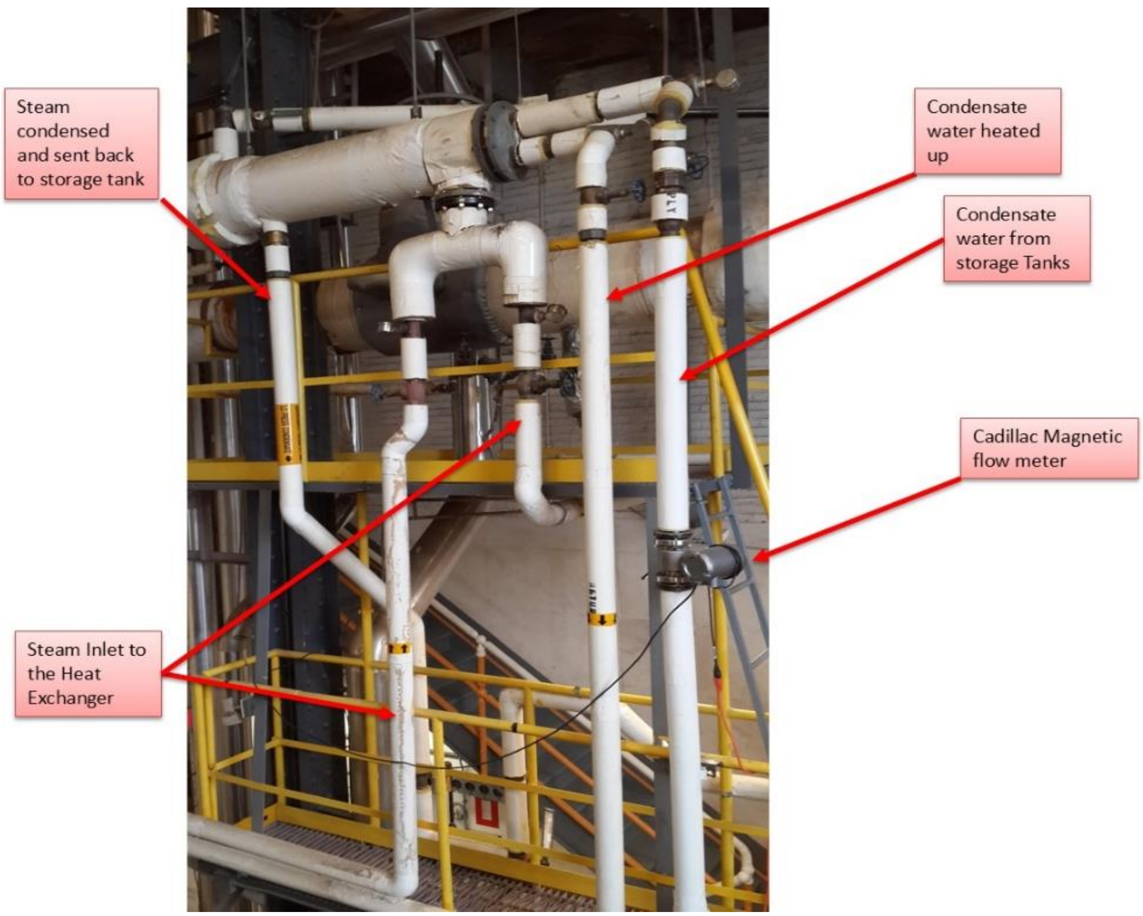


Figure 21: Vent condenser assembly

1.3.1 Case 1

The variable speed pumps were configured to run in pressure control mode, i.e., the pumps were running to satisfy/achieve a constant pressure of 43 PSIG (this value was selected based on the steam power plant's staff), in order to emulate the work of constant speed pump. In this case, both pumps were supplying condensate water for both the vent condenser and the deaerator (DA) tank. For this case, the total discharge flow was regulated for both types of pumps using the control valve. Therefore, the variable speed pump was working in the same mode as the constant speed pump.

1.3.2 Case 2

In this case, the variable speed pumps were running in level control mode, and trying to provide the required condensate water just to the DA tank based on the steam power plant demand. The valve in the 2" conduit line (Fig. 17) that was feeding the vent condenser with condensate water was closed. Therefore, the flow was limited only to the DA tank because the variable speed pumps were configured to provide the condensate water to the DA tank based on the water level sensor signal. The pumps were unable to provide the necessary pressure head to lift the condensate water to the vent condenser. The control valve in this case was fully open when the variable speed pumps were running, and the flow was regulated using the VFD controller. On the other hand, the constant speed pump was providing the condensate water to the DA tank in the steam power plant's basement. Thus, the constant speed pump's discharge flow was regulated to meet the steam power plant demand using the control valve. Even though the constant speed pump was able to provide the pressure head to feed the vent condenser, the vent condenser valve was shut-off. Therefore, the provided water was limited only to the DA tank in order to have the sane task for both types of pumps. Again in this case, the vent condenser was isolated from the steam power plant system.

However, in order to calculate the reclaimed energy from the vent condenser and the extra energy required by the constant speed pump to lift the water to the vent condenser, two approaches were used.

The first approach is calculate the energy gain from the vent condenser based on the water fed to the vent condenser [measured by the Cadillac magnetic flow meter] and the temperature rise across the vent condenser's inlet and outlet [measured by the temperature gauges installed in the water inlet pipe and water exit pipes] (see Fig. 21). Then, that calculated energy was converted into an equivalent amount of natural gas required to provide that amount of energy, based on the boiler efficiency, which was calculated from the information provided by the steam power plant log sheets (Appendix F). These pieces of information were the total steam generated, the temperature of the water inlet to the boiler, the temperature of the saturated steam, and the amount of natural gas consumed.

The second approach is to calculate the required power for the constant speed pump just to provide the necessary driving pressure to lift the water to the vent condenser. This method will be explained in more detail later. Furthermore, this approach will estimate the minimum required pressure that is necessary to lift the water to the vent condenser.

Two ONSET HOBO data acquisition units were used to gather data in this project. These types of data acquisition units offer a wide range of time intervals that can be set to log the data. The user can select the time interval that is more applicable for the unit for which the data was gathered. In this project, a one minute time interval was selected. One of the data acquisition units was installed next to the pumps (Worthington constant speed pump and the Grundfos variable speed pumps); and the other was installed next to the DA tank. Power consumption, flow rates, pipeline pressures, and pressure drop across the control valve were stored in the data acquisition units for selected time periods. Then using the HOBO software [already installed on the Gateway Notebook], the logged data was transferred to the Notebook; and LCC analysis was performed. After reading all of the data logged by the data acquisition units, the data acquisition units were restarted in order to be ready for storing new data. However, this process was not used for the Grundfos pumps, because the company has its own software, PC Tools E-Products software, to use in downloading all of the data related to the Grundfos pumps [10]. The time interval is selected by the software, and it is not constant; but the software logs the information whenever there is any change in the logged information.

1.3.3 Life Cycle Cost Analysis

This research used the NIST LCCA program provided by the DOE named Building Life Cycle Cost Analysis 5 (BLCC5) that can be downloaded from the DOE website free of charge [36]. The program can provide a complete economic analysis which evaluates the most cost-effective method from multi-proposed alternatives. The program can accept up to 99 alternatives that can be input and evaluated instantaneously in order to find, among those alternatives, the lowest LCC. The program is applicable for different economic project sectors. Therefore, it can be used for Federal government projects that are classified under FEMP guidelines and private sector projects that are directly affected by taxes [29].

This program is especially helpful for energy- and water-related projects (e.g., selecting the cost-effective cooling and heating system that can be installed for air conditioning in specific buildings). The software employs a simple equation, Eq. (6) [29], to add all of the project costs after adjusting them with adequate discount factors.

$$LCC = I + Repl - Res + E + W + OM\&R \qquad (6)$$

The BLCC is compatible with ASTM (American Society for Testing and Materials) standards associated with building economics [29]. Therefore, the program was selected to perform the LCCA for this project in order to identify the most economical pumping systems, i.e, the constant speed pump or the variable speed pumps, that can be used for the University of Kansas' steam power plant applications.

Chapter Two

2.1 System Setup

All measuring devices were installed during a previous study conducted by Fabian Schmidt [10], except that the study did not take into account the condensate water flow rate provided by the pumps to the vent condensing heat exchanger. This is where the project of this thesis began.

Two Siemens electromagnetic flowmeters (Model 3100) were installed by Schmidt in the discharge lines of the pumps being compared in this project in order to measure the condensate water flow rate being supplied by the condensate pumps being compared. (The Siemens flow meter that was installed in the variable speed pumps' discharge line was not used, and all of the information related to the variable speed pumps was taken from the PC-Tools software that will be explained later.) In addition, the Siemens flowmeters had a Sitrans Mag 5000 transmitter that was used for interpreting the signals transmitted from the electromagnetic flow meters (see Appendix B). The Siemens Sitrans Mag 5000 has an alphanumeric display interface screen showing the flow rate meter readings. Also, the Siemens Sitrans Mag 5000 was used for processing and controlling the output signals (4-20 mA) that went to the data logger; and it has many other functions. The same types of flow meter and transmitter, but with a lower flowrate range (0-28 GPM), were also installed by Schmidt in the recirculation line (the purple line on the schematic in Fig. 17). This magnetic flow meter was used for measuring the flow rate of recirculating condensate water through the 1" bypass pipeline (the purple line in Fig. 17). For the current thesis project, a Cadillac Electromagnetic Meter (MAG meter) was installed in the 2" pipeline that supplies the vent condensing heat exchanger with condensate water in order to measure the rate of condensate water to the vent condenser, as shown in Fig. 21. Refer to the Fig. 22 schematic for a detailed wiring diagram of the magnetic flowmeter [37].

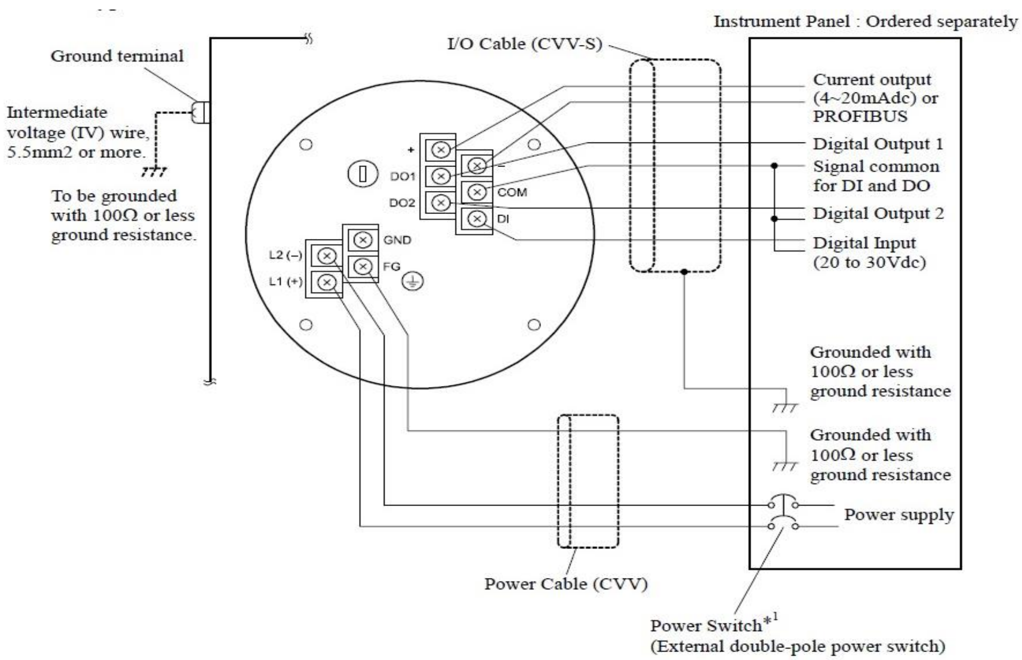


Figure 22: CMAG-II combined type converter wiring diagram (reproduced from Ref. 37)

There was a Danfoss MBS 3000 pressure transducer in the Grundfos variable speed pump discharge line (0- 145 PISG) installed by Schmidt [10]; and an OMEGA PX43E0-200GI pressure transducer was installed in the discharge line of the Worthington constant speed pump (0- 200 PSIG). Both pressure transducers measure the discharge gauge pressure delivered by the pumps. All measuring devices discussed above are shown in Fig. 17 and labeled in blue. The Danfoss MBS 3000 pressure transducer was directly connected to the control panel of the Grundfos variable speed pumps. The main function of the Danfoss pressure transducer was to make sure that the pumps were running at a set pressure point [input by the user] when these pumps were running in the discharge pressure control mode. The other function of this transducer was to acquire the water pressure data in the discharge line when the pumps were set to run in level control mode (see Fig. 17). Since both pumps take water from the same source (storage tanks), one Danfoss pressure transducer (0- 58 PSIG) was installed by Schmidt in the suction line of the Grundfos variable speed pumps in order to acquire the pressure on the suction side of the pumps with the assumption that both types of pumps have the same pressure at the inlet side due to the same water intake source [10].

One more Danfoss pressure transmitter (labeled in blue on Figure 17) was also mounted at the point just before the control valve where the DA tank is located in order to evaluate the pressure drop in the pipelines between the pumps' discharge points and the inlet right before the control valve [10]. A Grundfos Differential Pressure Sensor DPI, 0-2.5 bar, was installed to provide the pressure drop across the control valve [10]. Veris Power Monitoring H8044-0100-2 current transducers were connected to the Worthington constant speed pump control box in order to continuously measure the current that flowed in each phase and voltage values for determining actual power consumption of the pump [10].

As for the variable speed pumps, Grundfos has a built-in measuring device in the controller box. This measuring device automatically reads the true power consumption of each pump and stores it in the control panel's built-in memory. In order to run the variable speed pumps in level control mode, a SureSite visual indicator and level transmitter were installed by Schmidt [10]. The level transmitter's signal (4-20 mA) was fed to the Grundfos pumps' control panel via Belden 1120A 18 gage cables. An external power loop (12-24 DC voltage) was required to power all of the measuring devices with the exception of the magnetic flow meters, i.e., Siemens and Cadillac magnetic flow meters, which required 120V AC. Two Mastech DC Power Supply devices (HY3003D) were used to provide 10-24 DC voltage. One of them was placed next to the pumps in order to power all measuring devices nearby; and the other was placed beside the DA tank in order to power the remaining measuring devices. The measuring devices were wired using Belden

1120A 18 gage cables [10]; and, in order to direct the output signals from these measuring devices to the data acquisition units, HOBO H12-006 Data Loggers, 4-20 mA wires were used. Figure 23 gives a simple wiring diagram that shows how to wire one pressure transducer [38]. For the HOBO H12-006 data loggers, each data logger has four external channels with a total measuring capacity up to 43,000 readings [39].

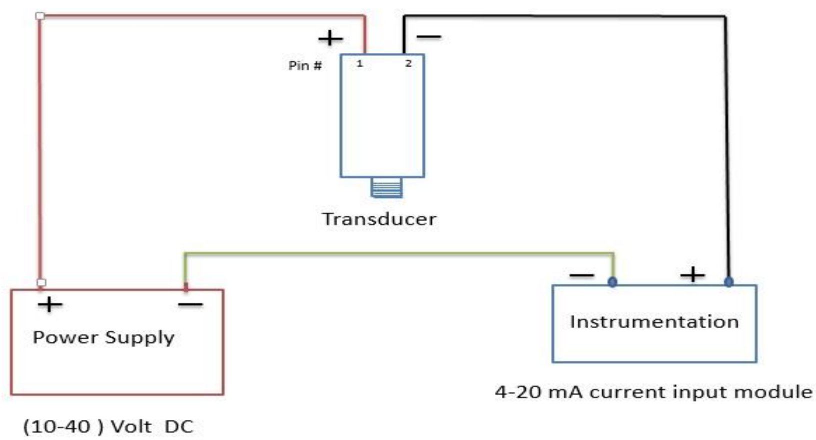


Figure 23: Pressure transducer wiring diagram (reproduced from Ref. 38)

The data was read from the data logger using a USB interface. In the Gateway laptop, the HOBO software was installed and set for each measuring device. The settings were configured by the user for each sensor. All sensors have a high and low measuring limit. So it is important to refer to the documents provided by the supplier for these values because 4 mA stands for the minimum value in the scaled column shown in Fig. 24 as Value 1, and 20 mA stands for the maximum value shown in Fig. 24 as Value 2. That scaling can be configured in the "Linear Scaling Assistance" window within the HOBO software, shown in Fig. 24.

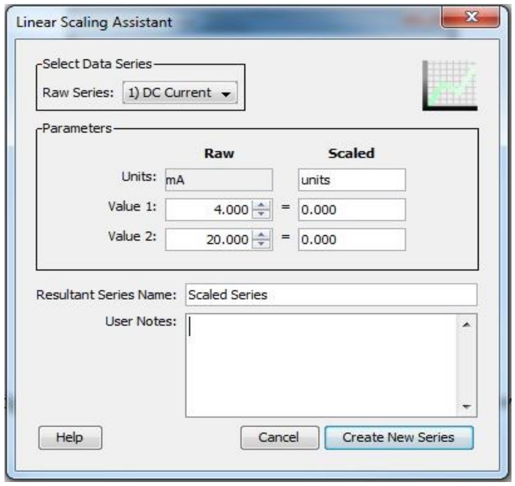


Figure 24: Screen shot of linear scaling assistant window

After these values are configured, HOBO creates a linear relationship between the raw values and their corresponding scaled values. This linear scaling relationship was used to convert the signal coming from the sensor (4-20 mA) to a resulting measurement, depending on the type of sensor. Appendix B shows all detailed technical specifications and pictures of the measuring equipment: Cadillac magnetic flow meter, Siemens magnetic flow meters, Omega magnetic flow meter, data logger, pressure transducers, level sensor, power supply, and power monitor sensor.

2.2 Project Troubleshooting

While working on gathering data for this project, many problems were encountered and addressed. For future study, it is important to detail these difficulties for the purpose of saving future students' time and effort. In order to measure condensate water flow rate to the vent condenser, a decision was made to install a flow meter in the 2" pipeline represented by the orange line that supplies the heat exchanger [vent condenser] with condensate water (see Fig. 17). Therefore, a Signet 2536 paddle wheel flow meter [40] with an error of ±1% of its maximum range (0.3-20 ft/s) at 25 ^{o}C was purchased and installed by Sam Willoughby [41] in the location labeled in blue on the schematic in Fig. 17. Installing the paddle wheel flow meter was a project completed by Sam Willoughby in August of 2013. It can be seen that the reading range is given in ft/s. That is because these types of flow meters measure the flow velocity. Depending upon the internal pipe diameter in which the flow meter is installed, the actual flow rate can be calculated by simply using a K-factor which is defined as the “number of pulses a sensor will generate for each engineering unit of fluid that passes the sensor” [40]. Knowing the pipe diameter, material, and pipe schedule helps the user to identify the correct K-factor for the flow meter. After obtain the K-factor from tables provide by the manufacturer (see page 6 of Ref. [40]), the user can determine flowrate with data acquisition software such as the HOBO software of this project. The flowmeter produced a square-wave frequency output, i.e., digital pulses, not 4-20 mA.

After having the paddle wheel flow meter installed and wired by Sam Willoughby (see Fig. 25); it was found out that there were no output signals (pulses), and all data logger-stored readings were zeroes. The first possibility was that the data logger was not functioning. Therefore, an oscilloscope was used to see if output pulses were being produced. However, even when using the oscilloscope, no pulses were found. Then a decision was made to remove the flow meter and check its components in order to make sure that nothing was preventing the paddle wheel from spinning. These types of flow meters are not recommended to work with a fluid containing particles or impurities, which the steam power plant has. After taking the flow meter out, it was found out that corrosion particles had collected on and around the paddle wheel, preventing it from spinning. The reason behind the metallic corrosion particles being attracted to the paddle wheel was that most paddle wheel flow sensors have magnets embedded in each paddle’s blades. These magnets are used to produce the output pulses [42]. Figure 25 shows the corrosion built up on the flow meter’s blades.

A. Installing the paddle wheel flow meter

B. Removing the paddle wheel flow meter

Figure 25: Installing the paddle wheel flow meter (A) and removing it (B) due to the built up corrosion inside

The paddle wheel flow meter was chosen over other flowmeters due to its low cost [41]. However, after removing the paddle wheel flow meter, it was decided not to reuse it in the steam power plant project because of the delay that the project had while dealing with this problem. The other reason behind not using the paddle wheel flow meter was that it requires a uniform flow profile across the pipe in order to have an accurate measurement. Therefore, it is not preferred to have fittings before and after the flowmeter for specific lengths of straight pipe. For example, if a valve is present the upstream of the paddle wheel flowmeter, which was the same case in the steam power plant application, it is recommended by the manufacturer to select a location with a straight pipe length of 50*I.D. For this case, that was 100” on the upstream side of the 2” pipe. Moreover, there was a straight pipe length of 5*I.D. that was 10” of straight pipe on the downstream side of the paddle wheel flow meter, in order not to have swirling flow [40]. In place of the paddle wheel flow meter, a Cadillac Electromagnetic Meter [43] from an old experiment was used and installed to provide the necessary data.

Since the Cadillac Electromagnetic Meter (MAG meter) had been obtained seven years earlier, calibration was needed before installing it in the steam power plant. Even though the magnetic flow meter was factory calibrated and had been used for an earlier experiment, its ability to work accurately needed to be verified. In order to check the functionality and accuracy of the seven-year old magnetic flow meter, a decision was made to build a flow meter testing system for the purpose of calibrating and testing the Cadillac magnetic flow meter, with the setup being able to perform other tests required in the future. The setup was simple. One pump and two flow meters on a mobile system (see Fig. 26).

The pump used was a constant speed Grundfos pump CR20-16, with a nominal flow range from 25 to 70 GPM, which was able to produce approximately 770 ft of head, or approximately 333 PSIG, at 68 ^{o}F [3] for nearly no flow [“Dead Head” or “static head”]. The motor attached to the CR 10-16 pump was a 15 HP, 460 V, 3-phase motor. The net weight of the pump along with the motor was 418 lb. See Appendix A3 for pump details. The CR 10-16 pump was connected to a 2" PVC (schedule 40) plastic pipe on the low pressure side and a 1.5" thick wall dark gray PVC pipe (schedule 80) on the high pressure side. The pump was fed with water from a 100 gallon High Density Polyethylene (HDPE) tank having dimensions of 38"L x 30"W x 26"H. A bulkhead 2" fitting connected to bottom of the tank, then 2" pipe extended down from the bulkhead to the pump intake side (suction side). On the intake side, around 5 ft of PVC (schedule 40) clear plastic pipe was placed right after the pipe that extended down from the tank (see Fig. 26). The decision to use the clear pipe was made so that an operator could see through the pipe in order to insure that there were no air pockets or bubbles developing in the pump's inlet.

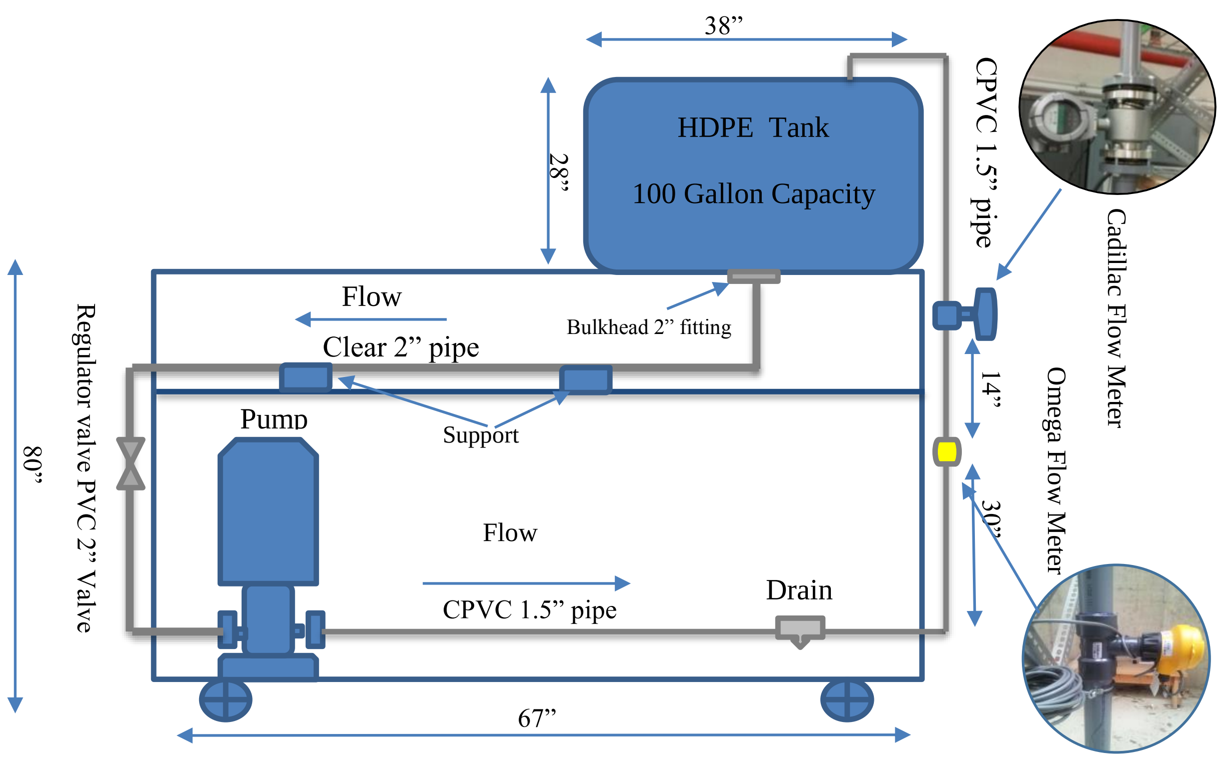


Figure 26: Apparatus schematic

It is not desirable to have air bubbles in the flow stream, because if they move across the magnetic flow meter's electrodes, the output signals will become unstable, especially when these bubbles contact the magnetic electrodes [44]. In addition, air bubbles may cause serious damage to the pump's impeller.

In order to regulate the flow through the testing loop to a desired value, a low-pressure CPVC ball valve [2" unthreaded socket end] was placed in the upstream line of the pump. The valve also was used to shut off the water flow whenever test apparatus equipment was not in use. In any recirculation loop, there must be a drain installed at the lowest point in the system so that the recirculation line and the tank can be drained (see Fig. 27). Then, when testing is not being performed, the system needs to be stored. For these reasons, the drain consisted of: an inline reducing tee (schedule 80), a (1-1/2x 3/4x1-1/2 Pipe Size) CPVC fitting, a pipe nipple PVC (3/4 pipe size -2" length), a CPVC ball valve 3/4" NPT female connection, a brass barbed hose fitting (3/4" Hose ID x 3/4" NPTF Male Pipe), and PVC tubing (3/4" ID, 1" OD, 1/8" Wall Thickness, 5 ft length). Figure 27 shows the drain location and a completed assembly.

Figure 27: Drain assembly of the testing system

For the testing system, there had to be two flowmeters. At least one of them must be calibrated while the other needed to have highly accurate measuring capability that can be counted on to calibrate the other flowmeter [whose accuracy is to be determined]. An Omega FMG3002-PP magnetic flow meter was employed in the system. The flow range of the FMG3002 flow meter was (0.15 to 16.4 ft/s) with an error of ±0.5% of its reading @ 25°C (77°F). The omega magnetic flow meter could be installed in wide range of pipe diameters from (0.5” to 12"). For more information on the Omega flow meter, see Appendix B8. However, for each pipe material and size, a specific fitting was required to accommodate the FMG3002-PP magnetic meter, along with known maximum and minimum reading values to be used for scaling (see Fig. 24). For each pipe diameter, there is a different maximum reading limit. For example, if the Omega magnetic flow meter was installed in CPVC 1.5” pipe (schedule 80), the Omega magnetic flow meter can read a flow rate up to 90.52 GPM. However, if the same Omega magnetic flow meter were installed in CPVC 2” pipe (schedule 80), it can read a flow rate up to 155.53 GPM [44].

A K-factor is not applicable in this type of flow meter because of the different output signal type. The Omega meter’s output signal was a 4 to 20 mA current signal, the same output signal as the other measurement instruments in the power plant. Thus, another data acquisition system was not needed to read the 4-20 mA signals. The Omega meter was powered by an Agilent E3630A triple-output power supply. See Fig. 28 which shows the magnetic meter installed. Figure 29 shows the magnetic flow meter connection schematic [44].

Figure 28: Omega FMG3002 magnetic flow meter installed in the testing system

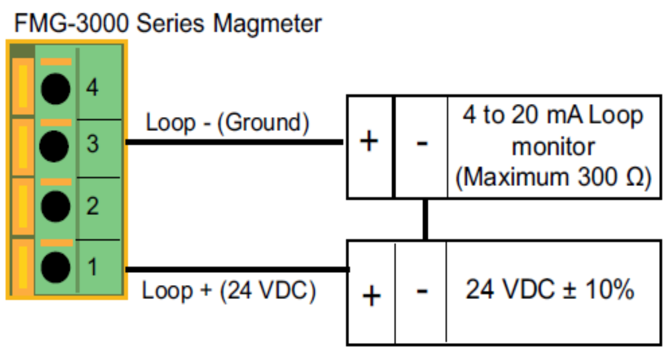


Figure 29: FMG-3002-PP magnetic flow meter connection schematic (reproduced directly from Ref. 45)

For accurate measurement, it was recommended by the manufacturer to have the magnetic flow meter installed in a location where there is an adequate length of straight full pipe directly upstream/downstream of the magnetic flow meter, in order to insure fully developed turbulent flow. There are different upstream/downstream pipe lengths recommended by the manufacturer, based upon different fitting connections. Taking that into consideration when installing the Omega flow meter in the testing system discharge loop which had 1.5" diameter piping, a distance of 30" (20*I.D. recommended) was selected for the magnetic flow meter upstream, and a distance of 14"

(10*I.D. recommended) was selected for the Omega flow meter downstream. The 10*I.D. distance was selected because of the Cadillac magnetic flow meter was next to the Omega. In order to ensure that flow was fully developed when the flow left the Omega meter, the distance after the Omega was selected to be 10*I.D. However, the available distance was 14” instead of 15”. This left the distance 1” short on the downstream side. However, according to the Omega manual, only 7.5” (5*I.D.) was recommended downstream of the flow meter. But in this case, the other magnetic meter (Cadillac) was present downstream of the Omega meter as discussed earlier. Therefore, to have an accurate measurement, an extra straight pipe length of 7.5” (5*I.D.) was added to be 15” (10*I.D.) total. Thus, being 1” shorter was reasonable. This distance was less important because of two reason: 1) the installation used a rough factor of safety of two; and 2) the Cadillac flow meter, did not require 5*I.D. of upstream pipeline, but only required 2.25” (1.5*D) [45].

The Cadillac flow meter, whose accuracy was being determined, was installed downstream of the Omega flow meter in the pump discharge line using two flanges (CPVC 1.5” pipe, schedule 80). See Fig. 26 for pictures and the apparatus schematic.

In order to support the pipelines on the discharge line, and provide vibration damping when the pump was running, rubber-cushioned loop clamps were used to immobilize the discharge line as shown in Fig. 30.

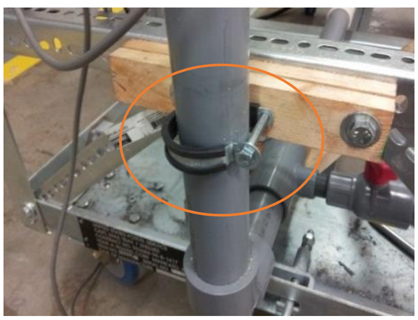
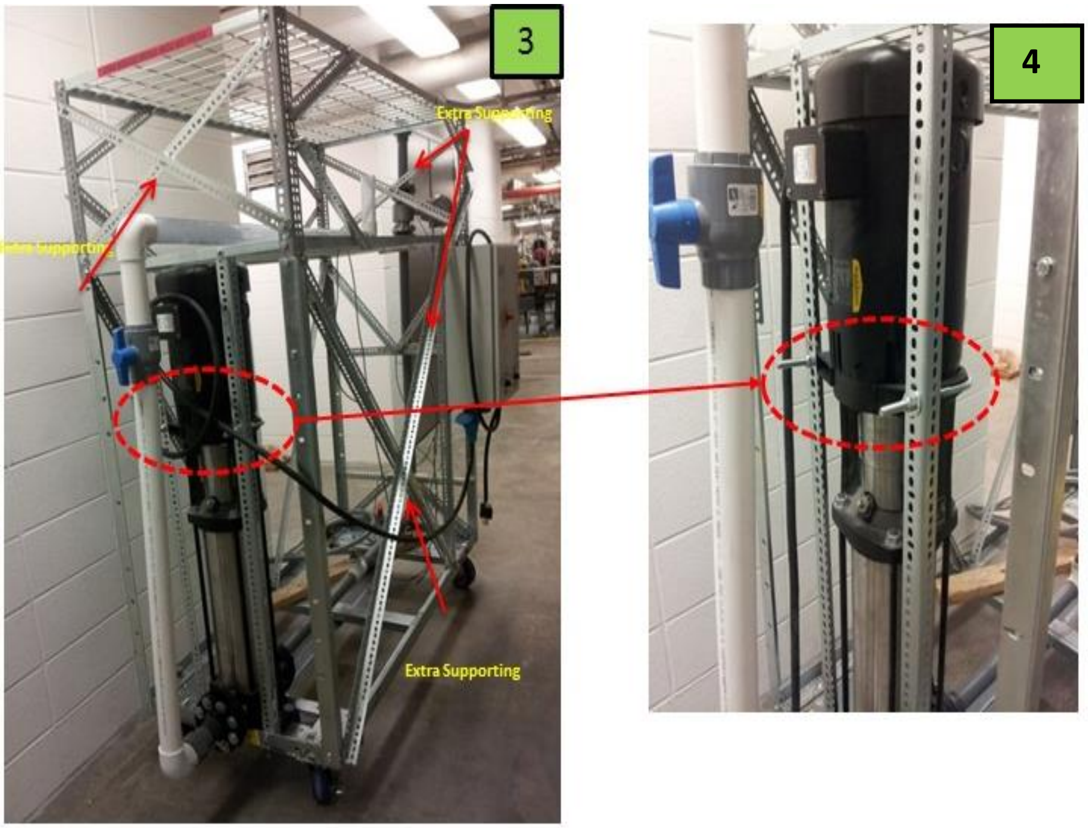


Figure 30: Pipe clamp for holding the discharge pipeline

All of the pipes on the discharge side of the pump were 1.5” (schedule 80) Chlorinated Poly Vinyl Chloride pipe (CPVC) that can withstand pressures up to 470 PSI at 73 ^{o}C [46]. Even though the maximum pressure that the pump can produce is 333 PSIG, as discussed earlier, a safety factor of approximately 1.4 used. The maximum pump pressure of 333 PSIG can be produced only for the case when there is no flow, i.e., “dead head.” The dead head situation should not occur with the flow testing system because the purpose of this testing system was to calibrate/test the two magnetic flow meters when there was flow. Moreover, the only regulating valve was installed on the suction side. Thus, accidental valve closure would not cause a dead head situation. However, the testing system was designed to handle much higher allowable working conditions in case this system were employed for tests requiring the 333 PSIG. On the other hand, the CR10-16 pump discharge diameter was 2”, but the Cadillac flow meter diameter was 1.5”. Therefore, in order to avoid having a reduction fitting before the Cadillac meter, and to fulfil the straight pipe length requirements within the structure’s limited dimensions (see Fig. 26), all of the pipes on the discharge side were selected to be 1.5” diameter. The reduction was placed right after the pipe-pump connection, in order to have fully developed turbulent flow on the discharge side.

The frame within which all of the pipelines (discharge and suction), pump, and the tank were placed had dimensions of 67” L x 30” W x 80”H (see Fig. 26). In order to well-support the frame for handling pump vibration while it was running, as well as holding the heavy tank, extra supports were used to reinforce the structure. Therefore, trusses were placed on the structure’s corners. For supporting the pump when it was running, a zinc-plated steel 6-5/8" OD vibration–dampening U-bolt was used [this U-bolt was used for holding a 6” pipe diameter, and reduce pipe vibration with a thermoplastic elastomer cushion]. In order to have better pump vibration control, this U-bolt was used, and modifications were made in the Engineering Shop at the University of Kansas, Lawrence campus (see Fig. 32b). The modification was that, because the base-pump's diameter was more than 6", an extra extension in length had to perform on the U-bolt. The extension represented by welding an extra 5/8" fully threaded shaft length to extend the legs of the U-bolt. Figure 31 shows the steps of building the structure, the structure's trusses and the pump holding assembly. In order to connect the trusses to the main frame, holes needed to be drilled, where the frame surface had no holes for 3/8" bolts. These were used to connect the trusses to the main frame structure. See Fig. 32b.

31(a) Placing the pump on the structure



31(b) Placing trusses and the pump holding assembly

Figure 31: Building the frame, extra support for frame and the pump holding assembly

All of the drilling was carried out manually. For this reason, building the setup took four more months than the one month that was scheduled.

After connecting all of the pipes and the fittings using plastic pipe cement and cleaner, the main tank was placed and connected to the suction pipe using a 2” union. Then, the calibration was carried out for the Cadillac magnetic flow meter. See Appendix C1 for calibration results. The Cadillac flow meter showed reasonable agreement with the Omega flow meter. The Cadillac flow meter reading was lower than the Omega flow rate reading with an overall error 5.33%. This work, from purchasing all of the required materials for the setup, building the setup, performing the calibration, through having the Cadillac magnetic flow meter installed, delayed the project five months.

After having the calibrated Cadillac flow meter installed in the 2” pipeline at the KU power plant (see Fig. 17 and Fig. 21 for Cadillac flow meter location), one of the Siemens magnetic flow meters installed in the discharge line of the Worthington constant speed pump was found to have faulty readings. The readings were less than these of the Cadillac flow meter. In addition, the pressure transducer (Danfoss MBS 3000) installed on the discharge line of the Worthington constant speed pump was showing negative readings. All recorded data were negative values, clearly impossible to have a vacuum pressure in the discharge line of that pump.

It was observed that the Siemens flow meter was reading 30.5 GPM (30 minute average of recorded data), while the Cadillac flow meter was reading 93 GPM (35 minute average of recorded data). By close examination of Fig. 17, this discrepancy was obvious because it can be seen that the Siemens flow meter reading should be higher than that of the Cadillac flow meter. This is because the Siemens flow meter was installed in the total discharge line, while the Cadillac flow meter was installed in a branch line. Consequently, the Siemens flow meter was determined to be faulty.

As for the faulty pressure transducer, it was replaced with a new one that had been kept from the previous study [10]. The new pressure transducer had the same specifications (Danfoss MBS 3000), able to read pressure in the range of 0-4 bar; and its output signal was 4-20 mA. After testing the sensor for functionality, no data was taken from it after that test because all of the focus was on calibrating the faulty Siemens flow meter. There was no point in taking pressure data while the Siemens flow meter in that same discharge line was faulty.

After checking the wiring and testing the output signal using a multimeter, it was found that the Siemens flow meter had no problem with the wiring; but the output signal amplitude was comparatively low. Also, the flow rate readings of the two Siemens meters installed in the discharge lines of the constant and the variable speed pumps (see Fig. 17 for more information) were compared. The Siemens flow meter installed in the variable speed pump line was showing a flow rate of 147 GPM, while the Siemens meter installed in the discharge line of the Worthington pump showed a flow rate 23 GPM under similar operating conditions. Therefore, the faulty Siemens magnetic flow meter had to be removed in order to calibrate it.

Before scheduling a time with the steam power plant staff to have the faulty Siemens flow meter removed, the calibration system had to be modified because the calibration system was built to handle the 1.5" diameter Cadillac flow meter, not the 4" Siemens flow meter. Thus, the pipelines in the calibration system had to be changed in order to accommodate the larger diameter of the Siemens flow meter, considering time and costs factors.

Three models were presented and evaluated in order to select the most reliable design and most reasonable cost to carry out the Siemens flow meter calibration. The selected design was one which just required an extension of the previous calibration system in order to hold and carry the relatively heavy pipes (4" pipelines filled with water). The new extended calibration system utilized all of the pipes and fittings on the intake side, and almost 70% of the pipes and fittings on the discharge line were reused. Some of the fittings could not be utilized in the new expanded calibration system such as the two flanges (CPVC 1.5" pipe size, schedule 80), the 1.5" CPVC 50" pipe length (schedule 80), three 1.5" CPVC elbows (schedule 80), one union, and the whole drain assembly.

In the selected design, pressure-treated plywood (4ft x 6 ft) was used and attached to the old setup by two 3/8” bolts in order to hold the extended pipes. The reason behind building the expansion, and not using the old calibration setup’s whole structure was because of the straight pipe length requirements by the Siemens flow meter’s manufacturer. In order to install the Omega magnetic flow meter in the discharge line in which the pump was running, the manufacturer’s manual required a straight pipeline length of 75” (50*I.D.). However, that length was not available in the old setup because the old setup’s total length was 67”. Besides that, where the pump was placed in the setup, it occupied a length of 20” out of that total 67” setup length. Therefore, only 47” was available, which was not sufficient. Furthermore, the Siemens flow meter needed a straight pipe length on the upstream side of 20” (5*I.D.) Therefore, expanding the system was inevitable. Plywood was used so that each of the required discharge pipeline lengths, i.e., the 1.5” pipeline and the 4” pipeline, could be placed on and attached to, the plywood surface. This configuration was less expensive in effort and time than using racks to hold the discharge line. As just discussed, most piping and fittings were reused for the second calibration system. However, purchasing 4” pipe and fittings was unavoidable because the Siemens flow meter diameter was 4”. Therefore, 10’ of PVC unthreaded pipe, five 90^{o} degree PVC Pipe elbows, one 4” x 1.5” reducing coupling, and four PVC unthreaded 4” diameter flanges were purchased. All of the fittings and pipe were scheduled 80. Furthermore, rubber caster wheels having 220 lb weight capacity were also purchased to allow the new setup to move easily, and support the end of the plywood, that was not bolted to the frame.

In order to have a flexible design for reusing the Omega flow meter for any future needs (see Fig. 32). All of the fittings and required pipe lengths were connected together to make a one unit. This complete unit was attached with the two 1.5” threaded unions on the ends so that the unit could be removed easily from the calibration system (semi-permanent connection) and reconnected, instead of using glue connections which cannot be removed (permanent connection). Thus, the calibration system was ready for this calibration or any other future work required using the Omega flow meter (flexible usage). These two unions were also purchased (see Fig. 32). For more flexibility, the piping system was designed so that it could be disassembled into five section of reasonable size. These sections have been stored for future work (Fig. 33).

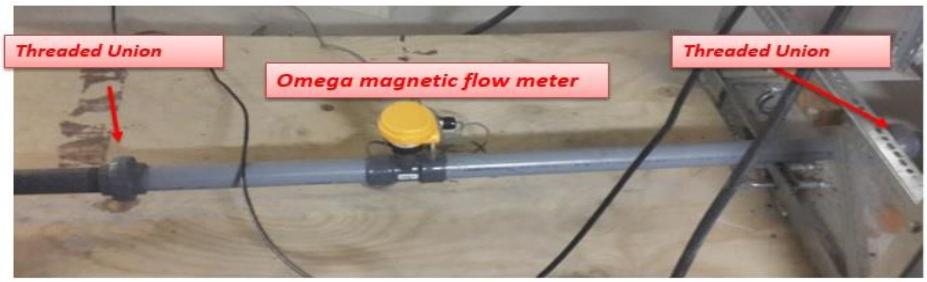


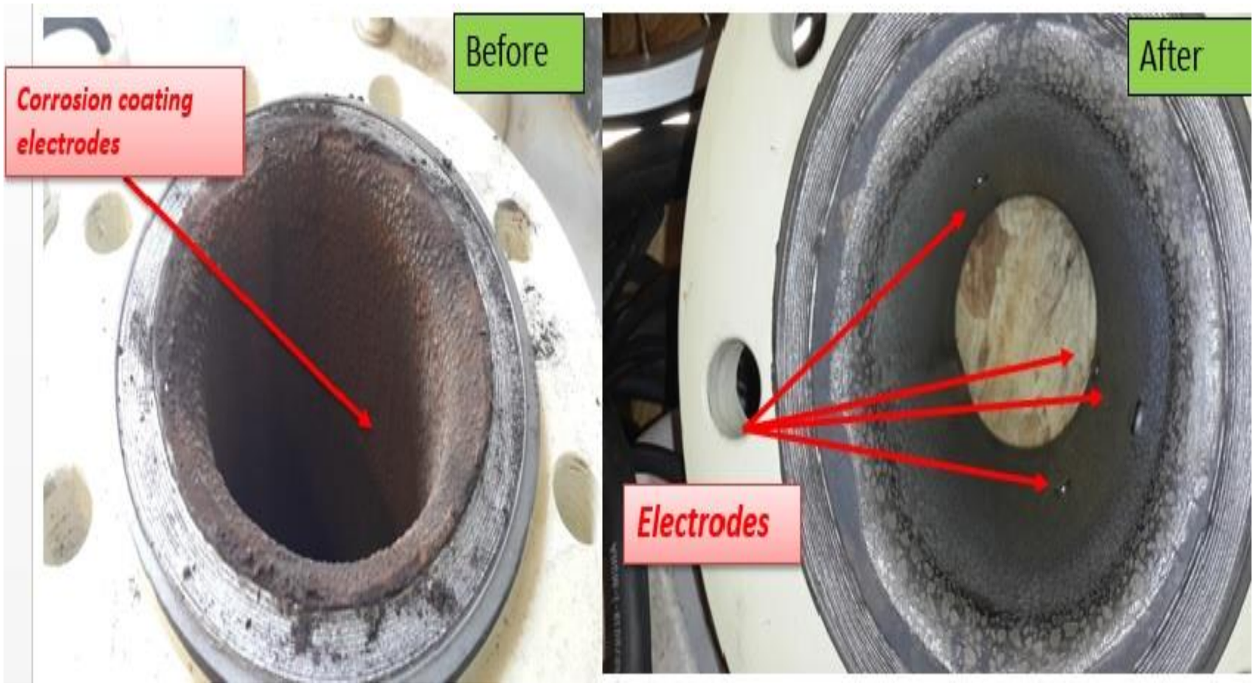
Figure 32: Omega magnetic flow meter

Figure 33: Expanded calibration system

After having the setup alterations finished, another challenge arose. When the Siemens flow meter was removed from the Worthington pump's discharge line, there were corrosion-particles coating the internal wetted parts of the faulty flow meter (see Fig. 34). That had coating caused the flow meter to read lower flow rates than it should have. The principle of a magnetic flow meter is derived from Faraday's Law which can be stated as "the voltage induced across any conductor as it moves at right angles through a magnetic field is proportional to the velocity of that conductor" [47]. In this case, the conductor is the water [or any liquid that is "electrically conductive"[47]] moving through the magnetic field produced by the flow meter across the pipe cross-section. Therefore, the voltage induced is the signal that is translated or converted to meaningful units of flow. However, the Siemens flow meter has an alternating current AC magnetic field. This type of magnetic flow meter is "highly sensitive to the coating on electrodes, since coatings cause a phase shift in the voltage signal" (see Ref. 47 for more information). That resulted in the reading error. After consulting with Siemens' technical assistance department, they recommended using scotch brite pads in order to clean the internal coated surface.

Having the faulty Siemens flow meter cleaned (see Fig. 34), it was ready to be calibrated using the modified calibration setup. See Appendix C2 for calibration results. After cleaning the Siemens magnetic flow meter, it was found that it worked accurately. However, the calibration was performed in order to verify that accuracy.

Even though the flow rate values of the other Siemens magnetic flow meter that was installed in the discharge line of the variable speed pumps were reasonable and acceptable as compared to the faulty flow meter, a decision was made to remove it from the discharge line, clean it using scotch brite pads, calibrate it, and reinstall it. See Appendix C3 for calibration results.



34a. Before 34b. After

Figure 34: Siemens magnetic flow meter before and after cleaning

There were two reasons for applying this process to the other Siemens flow meter. 1) The two Siemens flow meters were installed in the same working environment. Therefore, even though the variable speed pumps' flow meter had reasonably acceptable readings, that did not prove that the corrosion coating did not exist. The coating might have been in its first stages of development. 2) It was important to calibrate all of the present flow meters in the steam power plant using one reference flow meter, the Omega. It is noteworthy that the second Siemens flow meter had a thinner particle coating on the internal surface than the first Siemens flow meter - - for unknown reasons. Even though the second Siemens flow meter was calibrated and reinstalled in the discharge line of the variable speed pumps, it was not used to record these pumps' flow rate. The reason was that the PC-Tools software provided by Grundfos was able to record all system operating information such as pump flow rate, discharge pressure, and power consumption. Modifying the calibration setup, calibrating the two Siemens magnetic flow meters, and reinstalling them in the steam power plant delayed the project around two and a half months.

Having the two Siemens magnetic flow meters installed, and replacing the pressure transducer was not the end of new challenges in this project. The next challenge was that two pressure sensors, including the pressure transducer that was just replaced, were found to be giving odd readings. These were the discharge pressure sensor installed on the Worthington pump and the differential pressure sensor installed across the control valve. (See Fig. 17 for pressure sensors locations.) The new Danfoss MBS 3000 pressure transducer was also found to be malfunctioning. It gave zero readings shortly after replacing it (approximately one month from the replacement). The first assessment was that, because the water contained particles that were clearly seen on the Siemens' magnetic coating material, the nozzle of the Danfoss pressure transmitter was clogged. Therefore, there was no pressure transmitted to the pulse-snubber which is responsible for protecting and transferring the applied pressure to the transducer. See Fig. 35 for a schematic [48].

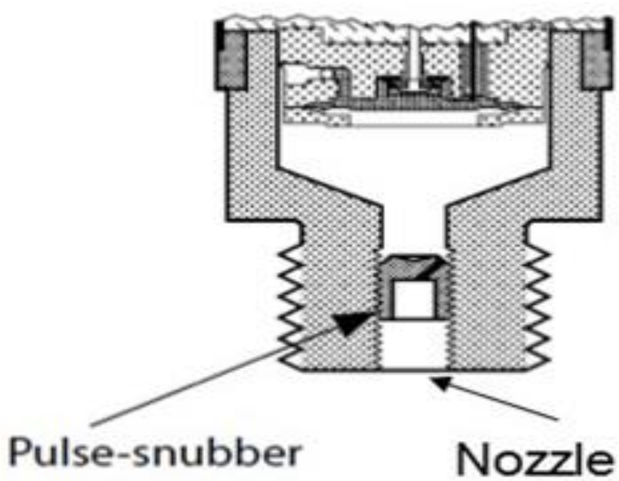


Figure 35: Danfoss pressure transducer (reproduced from Ref. 48)

Based on that assessment, the pressure sensor was removed; and it was found that there was material built up around the nozzle. After cleaning the pressure transducer (see Fig. 36), and reinstalling it back in the discharge line, the pressure transducer still did not work. The reasons behind the two malfunctioning pressure transducers was unknown until an observation was made when investigating the method by which the steam power plant staff was rotating the pumps' operation.

Figure 36: Before and after cleaning the Danfoss pressure transducer

As discussed in Section 1.4, there are four condensate pumps in the steam power plant. One of them is the variable speed pump set, while the others three are constant speed pumps. The steam power plant's staff normally rotated the operation of the pumps, with a different pump running each week. However, during the cold weather, the staff was running two pumps at the same time in order to meet the high demand. Consequently, the pressure increased in the discharge line in which the pressure transducer was installed. However, the pressure did not reach 50 PSIG even when two pumps were running. The maximum pressure that the pressure transducer could handle was 4 bar or (58 PSIG). However, if the staff were running two pumps, they started a third pump before shutting off one of the two running pumps. For instance, pump 1 and 2 were running for one week. In order to have pump 1 shut-off, the staff started a third pump from the two idle pumps, then shut-off pump 1. This pump shifting increased the pressure in the main discharge line up to 48 PSIG in cold weather. However, when pump 1 shut-off, the high pressure caused sudden closure of the check valve in the discharge line of pump (each pump has a check valve in its discharge line). This sudden closure resulted in the flow velocity decreasing while the pressure at the check valve increased. Thus, a pressure wave or a shock wave could develop because of the pressure increase at the check valve and travel up and down in the discharge line until completely stopped [49]. In conclusion, this pressure wave caused the damage to the pressure transducers. See Fig. 37 for a close up of the transducer. Therefore, a new pressure transducer (OMEGA PX43E0-200GI) was purchased to replace the Danfoss MBS 3000 pressure transducer.

Figure 37: Oil leaks from Danfoss MBS 3000

The new pressure transducer was selected to have a higher reading limit (0-200 PSIG) in order to handle the pressure increase in the discharge line when two pumps ran at the same time. In addition, it had a rugged flush diaphragm in order to work in a medium that included suspended particles.

Finally, the Grundfos Differential Pressure Sensor (DPI 0-2.5 bar) was also found to be faulty, with all of the readings being zero. After checking the wiring and measuring the output signal using a multimeter, there was no indication of any output signal from the sensor. Again, the first assessment was that material build-up had accumulated in the UNS 7/16” capillary tubes (the tubes that connect the sensor to the high pressure side and low pressure side). Therefore, there was no pressure transmitted from the pressurized pipe to the sensor due to water blocking the capillary tubes. The assessment was correct, and the capillary tubes were completely blocked. Thus, there was no water moving through the capillary tubes. See Fig. 38 for details. The capillary tubes were cleared using a pressurized air supplied from a compressor at the steam power plant. After cleaning the capillary tubes and the inlet ports of the sensor, the sensor was rewired and connected again. Subsequently, the sensor worked normally.

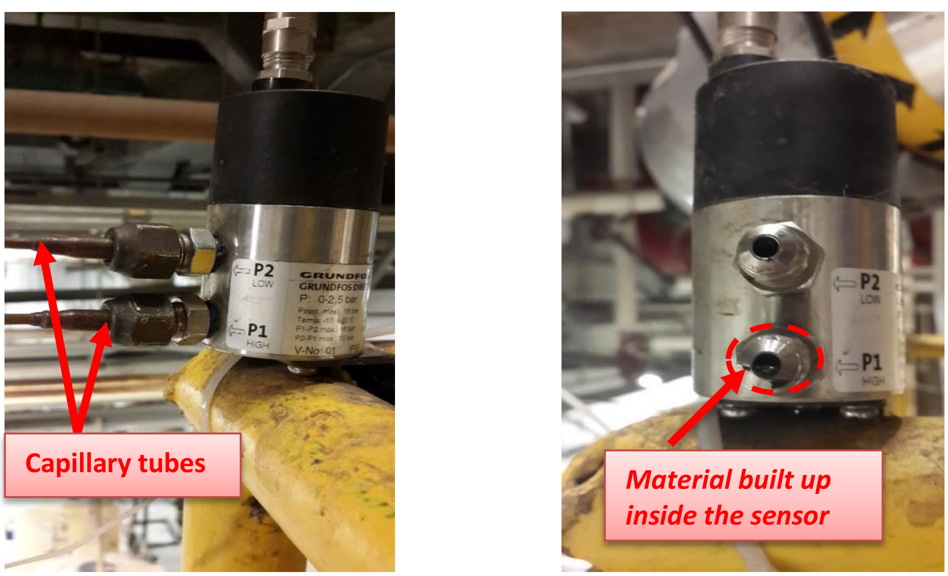


Figure 38: Differential pressure sensor DPI 0-2.5 bar

Once all of the measuring devices were ready, gathering the data for this project began in January of 2015. All measuring devices had been well-checked and calibrated.

Problems have been addressed during this project that were encountered during the previous study conducted by Fabian Schmidt [10], such as the variable speed pumps inability to provide the water for both the DA tank in the basement and the vent condenser on the first floor [see Fig. 17] when operating the variable speed pumps in level control mode. This inability can be explained by understanding the principle upon which the variable speed pumps operate (Affinity Laws) and the elevation at which the vent condenser was installed. According to the Affinity Laws, when the variable speed pumps run in level control mode at an arbitrary operating point, say 176 GPM, 100 ft of head, and a speed of 2984 rpm, if the pump's speed decreases by half, that shifts the operating point to 88 GPM, 25 ft of head, and a speed of 1492 rpm (see Eqs. (4) and (7)) [23]. One might notice that the pumps' head decreases by a factor of four when the speed drops by half.

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1} \tag{7}$$

All the terms used in Eq. (7) are defined in the Nomenclature.

The vent condenser required 40 ft of head just to overcome the static head. Therefore, the variable speed pumps in level control mode could not provide the vent condenser with condensate water due to low pressure head at low speed. Thus, whenever the variable speed pumps ran in level control mode, it was advisable to have the vent condenser valve closed in order not to overheat the vent condenser due to lack of condensate water flowing through its pipes.

One of the challenges found when running the variable speed pumps was that, once the set point (selected set point value by the user) was achieved or sometimes overachieved, the pumps' operation switched to minimum performance in order to save energy. Thus, one pump completely stopped, and the other pump ran at 30% of full speed. However, under these conditions, the pumps were unable to provide the needed flow to the DA tank. That caused the pressurized steam inside the DA tank to enter the discharge pipe lines which caused "steam hammering", wherein the pipelines made sounds similar to hammering on a pipe, causing the pipelines to vibrate. In order to avoid that problem, the minimum performance level was manually adjusted so that one pump ran at 42% of full speed and the second pump was off. For this arrangement, the pumps had the ability to continuously provide the needed water to the DA tank. Once that change to a 42% minimum was made, there was no steam hammering in the system. This change was not made randomly; but it was made after carefully examining the steam power plant load, DA tank pressure, and understanding the pump curve provided by Grundfos (Appendix A2). Therefore, it is very important to keep in mind that this change may not be suitable for all cases during the summer and winter. The user has to make that decision and understand at what percentage of full speed is needed in order to maintain the proper flow going to the DA tank for different times of the year.

Figure 39 shows the variable speed pumps' reaction when the set point was reached for the case when the minimum performance level was one pump running at 30% of full speed. The set point is the point at which the operator intends to operate the pumps. This set point can be a specific level in the DA tank or a specific pressure that the operator intends to make the pumps provide. From Fig. 39, it can be seen that the variable speed pumps hardly produced flow at minimum performance. In Fig. 40, when examining the variable pumps' speed during the time period of operation when there was no flow (see Fig. 39), one can see that the variable speed pumps were operating such that one of the variable speed pumps was completely stopped while the other pump was running at a speed ratio ($\alpha = n/n_{max}$) of 30%. Consequently, this minimum performance was found to be insufficient for this application profile.

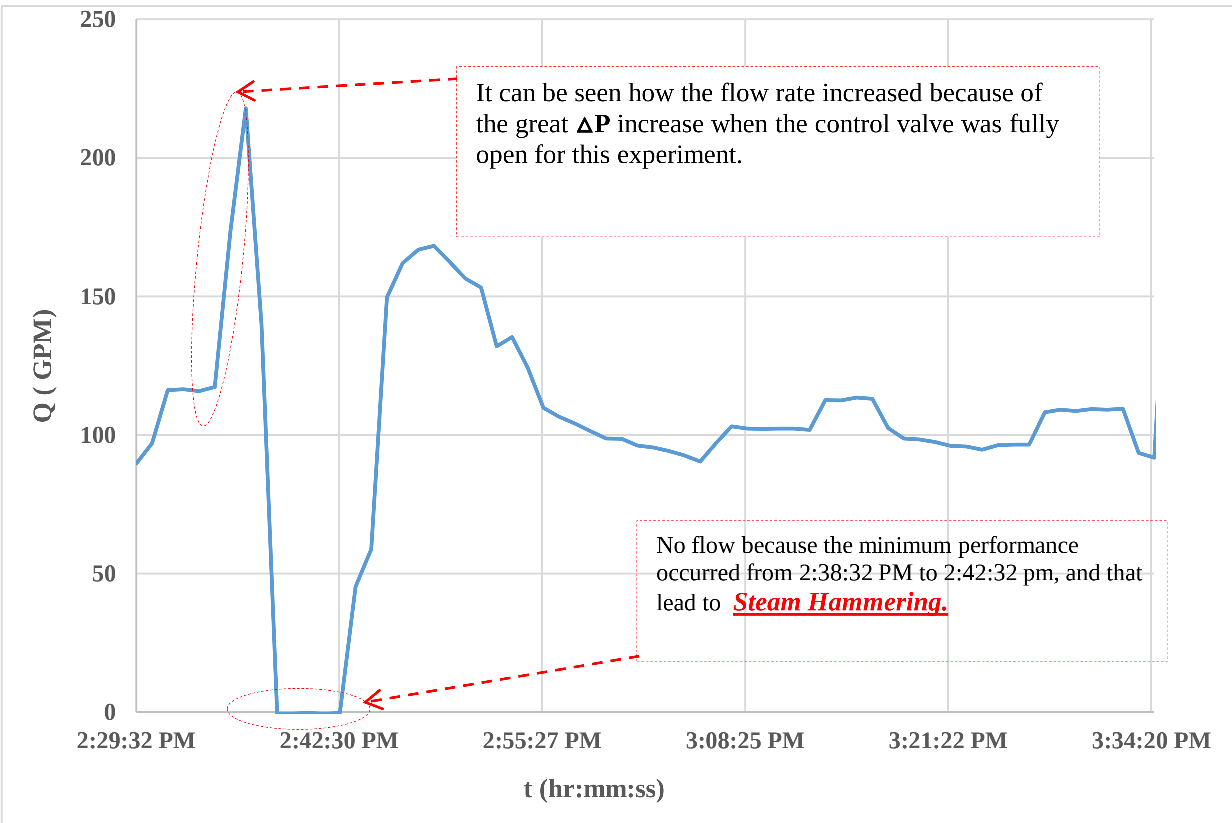


Figure 39: Grundfos pumps' flow rate curve on February 24, 2015

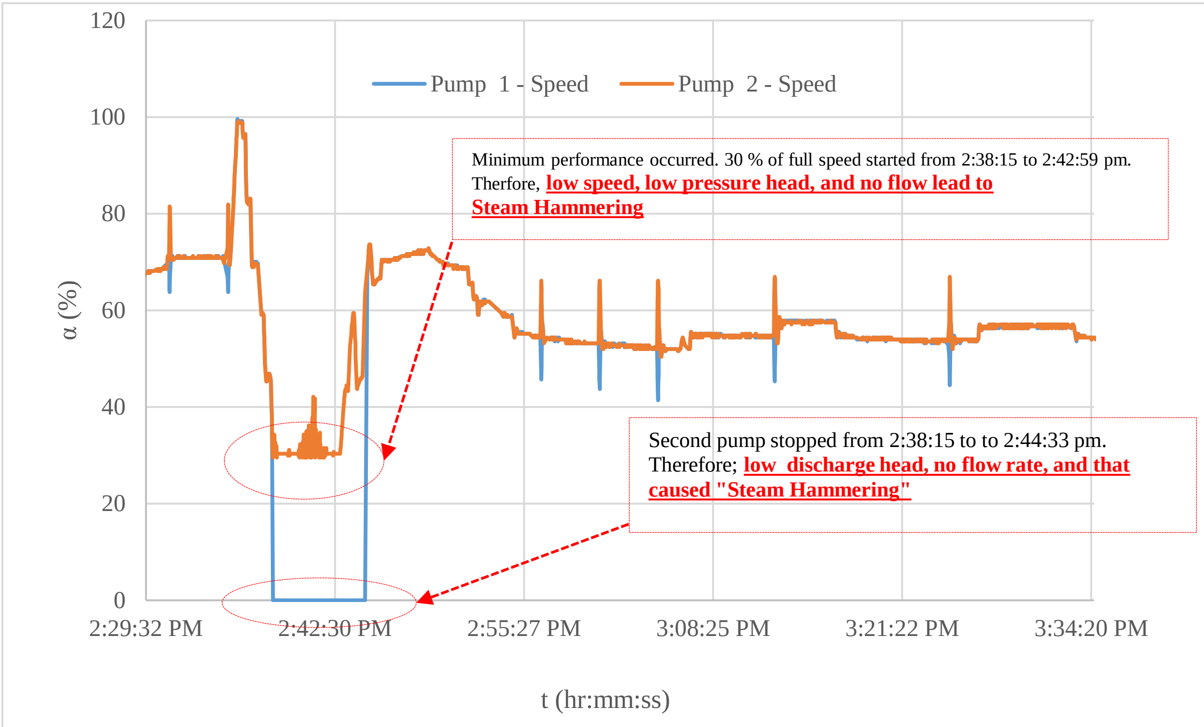


Figure 40: Grundfos pumps' speed ratio curve on February 24, 2015

Another difficulty encountered when operating the variable speed pumps in level control mode was that the pumps' sensitivity to a change in water level in the DA tank was extremely high. Whenever the water level decreased below the set point (52% of the full DA tank capacity), the pumps reacted in a very short time to compensate for that decrease. When the pumps achieved the set point, they slowed down in same manner. This variation resulted in exceeding the set point and never maintaining a fairly constant water level in the DA tank. Therefore, the pumps' sensitivity had to be adjusted [50]. Based on the idea of that the DA tank has a circular cross-section, and in order for the pumps to change the water level (e.g., from 51% to 52%), they needed some time to make that change gradually because the tank capacity at the center is higher than that near the top, or the bottom. Therefore, the pump sensitivity or integral time of the pumps' controller (T_i) was changed from 0.5 sec. to 2 sec. Figure 41 shows the results from changing the integral time to 2 sec. The integral time is the time set for the pumps controller to respond to any change occurs in the set point.

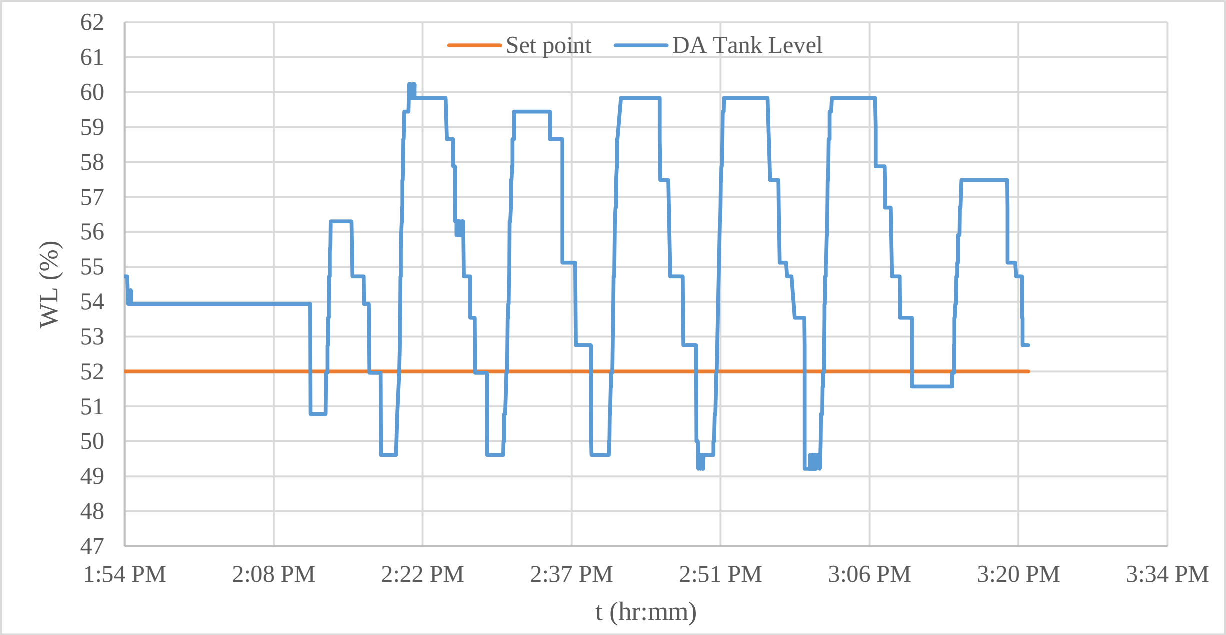


Figure 41: Water level in the DA tank vs. the set point while running Grundfos pumps on March 9, 2015

Again, changing the integral time (T_i) does not mean that value works for all situations when the variable speed pumps run in level control mode. That value depends on specific factors like the steam demand/load.

Even though the measured water level in the DA tank (WL (%)), as shown in Fig. 41, was not exactly the same as the set point (52%) most of the time, one might consider that the load/steam demand was also not constant and the steam demand changed often. Therefore, the variable speed pumps responded to changes in an attempt to maintain the water level at the set point required by the operator. Also, Figure 41 shows that the water level remained above the set point most of the time, because the pumps attempted to reach their maximum performance [the two pumps running at 100% full speed] in order to achieve the set point. Therefore, the flow rate peaked as shown in Fig. 42. These flow rate peaks caused the water level in the DA tank to be greater than the set point. However, when the pumps were signaled that the desired water level in the DA tank had been achieved or overachieved, the pumps lowered their speed; but by that time, the water level in the DA tank was already over the set point.

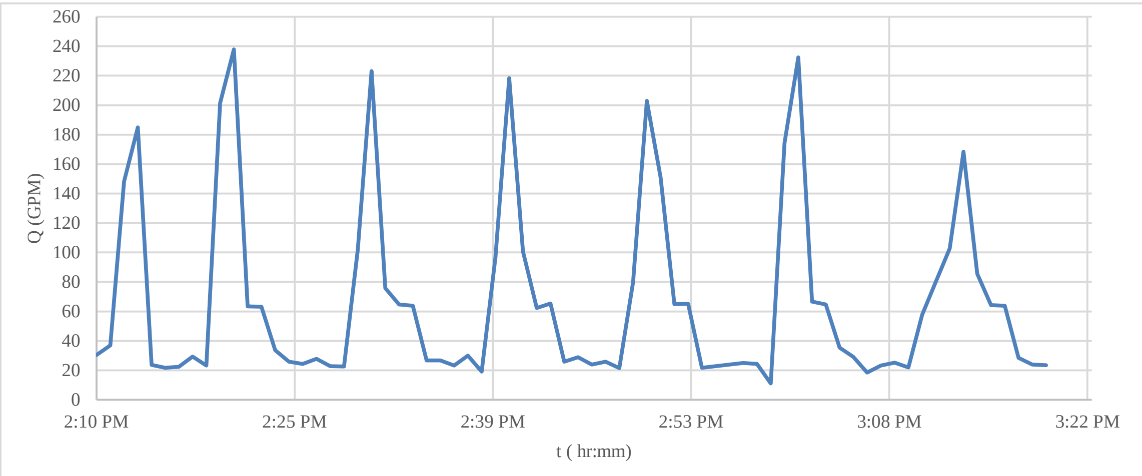


Figure 42: Grundfos pumps' discharge flow rate on March 9, 2015

Figure 42 shows high flow rate peaks, which reached 237 GPM, corresponding to the times when the water level in the DA tank was relatively high (60% full). On the contrary, corresponding to the times when the water level in the DA tank remained below the set point for relatively short periods, the flow rate was relatively low (20 GPM).

Finally, according to the Grundfos instruction manual, the operating mode should be changed from closed loop to open loop when the pumps were in level control mode. However, when the instructions manual was followed, it was found that the pumps did not react to the level control signals during operation, and the pump speed remained constant, even though the water level in the DA tank changed. Therefore, after many attempts, in order to make the pumps respond properly to water level changes in the DA tank, it was found that the pumps should be operated in the closed loop mode. This information was not found in the BoosterpaQ®- Hydro MPC instructions.

However, after contacting the Grundfos representatives [51], they advised that, when the operating mode was changed from closed loop to open loop, the pumps will not run according to the built-in algorithm configured by the manufacturer. Thus, keeping the system operating in the closed loop mode, and not changing to open loop, allowed the system to run normally and be most efficient.

When the set point was achieved or overachieved, the pumps automatically switched to minimum performance so that energy could be saved by this process, as explained previously. However, pump number 1was found to be always "on" during the operating time without switching the work to the pump number 2. Thus, there was no swapping duty each time the pump system worked at minimum performance. That mode of operation could cause pump number 1 to be overloaded when the system operated in level control mode. See Fig. 43 for more example data. This appeared to be a manufacturer's algorithm control problem, and the user could not make changes to modify this situation.

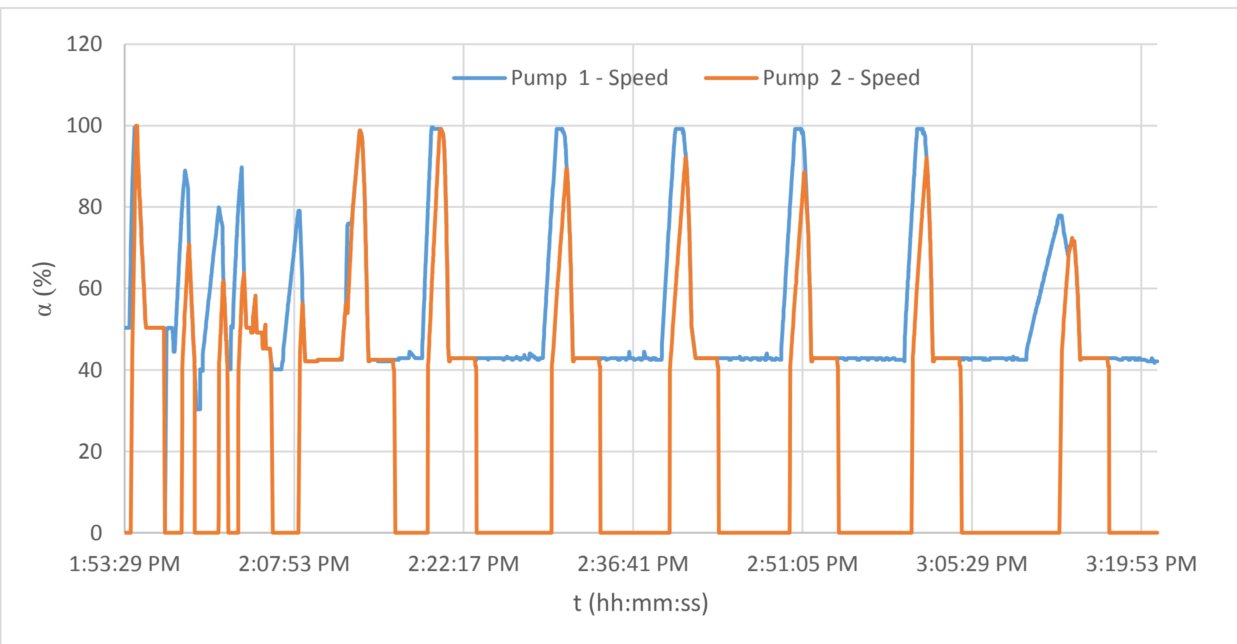


Figure 43: Grundfos pumps' speed ratio variation while in level control mode on March 9, 2015

The last problem found was related to the operating temperatures for the BoosterpaQ® CRE 15-3 variable speed pumps. From the pump curves, it was noticed that these pumps can be operated with the working fluid at a maximum temperature of 140 ^{o}F. However, this temperature limit might be exceeded in the KU steam power plant, because the condensate water in the storage tank might go up to 180 ^{o}F during summer time [30], and the average temperature of the condensate water in the storage tank then ranged from 150-160 ^{o}F. That led to the conclusion that the variable speed pumps were working in a harsh environment. After contacting the Grundfos representatives [51], they explained that the system can be operated for a rated temperature up to 176 ^{o}F based on sensor limitations.

2.3 Data Logging Procedure

The data recorded for this project followed a different scenario. Each of the two cases explained earlier followed a different pattern in gathering data. Therefore, it is more helpful to explain the scenario that was used for logging the data for each case. The data gathering was performed with an appreciable help from a graduate student Anurag Nanda.

2.3.1 Case 1

The data logged for Worthington constant speed pump and the Grundfos variable speed pumps covered three to five days for each type of pump at the time when the steam demand was at its maximum, i.e., winter time.

The data logged for each pump were power consumption, flow rate, discharge pressure, inlet pressure, pressure before the control valve, pressure drop across the control valve, variable speed pumps' speed ratio if needed, and the vent condenser flow rate. For the Worthington pump, the HOBO data acquisition logger was employed to log all of the listed information at one minute time intervals. However, all of the Grundfos pumps' related information was logged using the PC-Tools E-Products software provided by the Grundfos company, with the exception of recirculation water flow rate, if there was any, and inlet pressure, which were logged by the HOBO data acquisition logger at the time interval, i.e., one minute. The PC-Tools E-Products software was installed on a Gateway Netbook laptop. Therefore, in order to log the Grundfos pumps' data (i.e, power consumption, discharge flow rate, and discharge pressure), the Gateway Netbook laptop was left running in the steam power plant, and the software was restarted every day so that the logged data would not be lost for any reason.

The logged data for this case was for two months, February and March of 2015.

2.3.2 Case 2

In this case, the Grundfos pumps were running in level control mode, and they were configured to supply water just to the DA tank. Because of the pumps' low discharge head when they were running in level control mode, no flow was provided to the vent condenser. One should notice that, when in pressure control mode, the pumps can provide the water for both the DA tank and the vent condenser; but, for this case, the variable speed pumps' primary focus was to maintain the water level in the DA tank at a desired level or set point. That level was 52% of the full tank capacity, regardless of the amount of water flow to the vent condenser. Consequently, whenever the water level in the DA tank reached the desired set point, the variable speed pumps' reduced speed in order to save energy and not to over fill the DA tank, because the control valve was fully open. Thus, the pumps' discharge head significantly reduced whenever the pumps' speed became low. Making the variable speed pumps run in level control mode was challenging. Estimating the best minimum performance at which the pumps could run and the best integral time (T_i) for the pumps' sensitivity required the user to be familiar with the pumping system and the pressure drop in the lines. However, after changing the minimum pump performance (42% of full speed) and the integral time (see Section 2.2), the pumps were able to provide the condensate water to the DA tank without jeopardizing the DA tank and pipeline structure (i.e., without steam "hammering").

As discussed earlier, the vent condenser was isolated when running the variable speed pumps in level control mode. The same process was duplicated with the constant speed pump in order to have both types of pumps doing the same task for comparison. When dealing with the Worthington pump, all recorded data values over the total elapsed time were averaged arithmetically since there was no significant fluctuation in the data logged.

The data recoded for Case 1 was during the highest demand (winter time); and some data was recorded during moderate weather conditions in order to investigate any potential energy savings. Therefore, the data was gathered for this case in March and April of 2015 (on March 11, April 1, April 2, and April 9).

The data logged by PC-Tools E-Product software did not record the Grundfos pump information at a fixed time interval. The time spacing between the recorded data points was non-uniform because data acquisition was triggered by a change in values. For this reason, the Trapezoidal rule shown in Eq. (8) [52] was used to compute the average value for every recorded piece of information from PC-Tools software. After finding the area under the curve using the trapezoidal rule, that area was divided by the entire time span (t_j-t_o) during which the data was recorded in order to have the average value.

$$Q_T = \int_{t_o}^{t_j} Q(t)dt \approx \frac{1}{2}\sum_{i=1}^{i=j}(t_1 - t_0)\left[Q(t_o) + Q(t_1)\right] + (t_2 - t_1)[Q(t_2) + Q(t_1)] + \cdots + (t_j - t_{j-1})[Q(t_{j-1}) + Q(t_j))] \quad (8)$$

Figure 44 shows the terms used in Eq. (8).

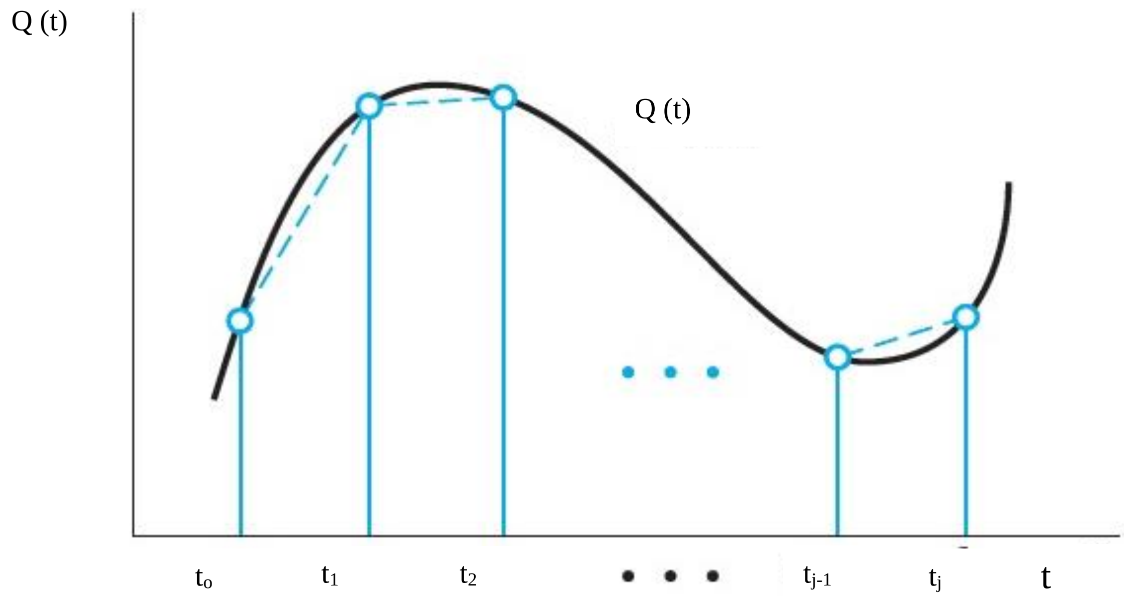


Figure 44: Trapezoidal rule (reproduced from Ref. 52)

2.3.2.1 Vent Condenser Calculations

As discussed earlier for Case 2, the vent condenser was isolated (not connected) when the Grundfos pumps were running in level control mode. Therefore, the steam power plant was not able to reclaim some of the energy from vented non-condensable gases. In addition, the steam power plant was not able to save the steam that typically condenses in the vent condenser. As for the amount of steam that can be reused from condensing the excess steam, most calculations neglect the total amount of bled steam (discussed in Section 1.3) that is often used to preheat the boiler feedwater in the DA tank. This is shown in the mass balance of Eq. (9) [53].

$$FW = MU + CR = S + BD \tag{9}$$

where FW= Feed water; Mu= Make-up; BD= Blowdown; S= Steam rate; and CR= Condensate return.

Equation (9) is used to estimate the make-up water that is necessary for replacing the lost water in a steam cycle. One could conclude that: if the entire amount of extracted steam is neglected in calculating the make-up water, then the excess steam that escapes from the DA tank will be even smaller. Therefore, this thesis will not investigate the amount of make-up water that was saved when the vent condenser was in use.

However, the reclaimed energy is the primary focus of this section. Therefore, approximate calculations are used to estimate the amount of energy that can be reclaimed from the vent condensing heat exchanger under discussion, which is [34]

$$E_{vent} = m^{o}_{water}\, cp_{water}\, \Delta T_{rise} \tag{10}$$

Then, in order to compute the amount of natural gas that would be needed to provide this energy without the vent condensing heat exchanger, the boiler efficiency was used, which is defined as [54]

$$\eta_{boiler} = \frac{(E_{steam} - E_{BFW})}{E_{Fuel}}\ 100 \tag{11}$$

The term "boiler" as used in this thesis, refers to the entire steam generator [boiler and the economizer].

or

$$\eta_{boiler} = \frac{m^{o}_{steam} h_{sat.steam} - m^{o}_{BFW} h_{sat.water(BFW)}}{m^{o}_{fuel}\,(LHV_{fuel})}\ 100 \tag{12a}$$

All terms in Eq. (12a) are defined in the Nomenclature.

The m^{o}_{steam} shown in Eq. (12a) refers to all generated steam from the boiler, which includes the amount of steam extracted from the main steam line and injected in to the DA tank in order to preheat the boiler feedwater in the DA tank (see Section 1.3). m^{o}_{BFW} can include the amount of blowdown water coming from the boiler, or it may not include that amount [54]. Consequently,

m^o_{BFW} was considered to be approximately equal to the amount of generated steam from the boiler, i.e., $m^o_{BFW} \approx m^o_{steam}$. As a result of this approximation, Eq. (12 a) can be re-written as

$$\eta_{boiler} = \frac{m^o_{steam}(h_{sat.steam} - h_{sat.water(BFW)})}{m^o_{fuel}\,(LHV_{\,fuel})}\ 100 \qquad (12b)$$

Since the fuel used in the steam power plant is natural gas, the Lower Heating Value (LHV) of natural gas was employed. That is equal to 1018.6 Btu/ft^3 [55].

The natural gas fuel flow rate in Eq. (12b) was provided by the steam power plant operating log sheets [Appendix F]. m^o_{steam} in Eq. (12b) was also given by the same log sheets [Appendix F]. The enthalpies for both the generated saturated steam ($h_{sat.\ steam}$) and the saturated boiler feedwater ($h_{sat.\ water(BFW)}$) were found from the steam tables [34] corresponding to the temperature and pressure of each. The BFW temperature was taken from averaging the maximum and minimum temperatures of the boiler feedwater flow. From observation of the boiler feedwater temperature in the hourly log sheets [these sheets were not attached to this document], the maximum and the minimum boiler feedwater temperatures were found to be 230 ^{o}F and 220 ^{o}F, respectively. Therefore, the average boiler feedwater temperature was taken as 225 ^{o}F. Having the BFW temperature, the enthalpy for that corresponding temperature was 193.3 Btu/lb$_m$. The generated steam enthalpy was taken from the fact that the all boilers were controlled to operate at a pressure of 170 PSIG. Therefore, from the steam tables for saturated steam at 170 PSIG, the enthalpy was taken as 1197.7 BTU/lb$_m$.

Finally, using this information in Eq. (12b), the daily boiler efficiency could be calculated. Even though the daily boiler efficiency information was given in the log sheets provided by the steam power plant, this document will not use that information because there was no explanation for how that boiler efficiency was calculated for those log sheets.

After calculating the daily boiler efficiency [Appendix F], the monthly average boiler efficiency was used to calculate the savings in natural gas from the vent condenser. Equation (12c) was used in order to calculate natural gas savings from using the vent condenser gain energy in the steam power plant. Equation (12c) is the result of combining Eq. (10) and Eq. (12b).

$$m^{o}_{fuel} = \frac{E_{vent}}{\eta_{boiler}\ (LHV_{fuel})}\quad 100 \qquad (12c)$$

See Appendix D for the monthly boiler efficiency and the energy gain calculations of the vent condenser.

As it can be seen from Eq. (12c), the output energy [the numerator] is the energy gain from the vent condenser, so that the Eq. (12c) can be used to predict the savings in equivalent amount of natural gas that would be burned by the boiler. Because the energy used to heat up the condensate water in the vent condenser comes from excess steam from the DA tank, the steam has the energy provided by the boiler, accounting for boiler efficiency. Therefore, the energy gain from the vent condenser simply represents the corresponding amount of natural gas that can be saved when having the vent condenser included in the system's operation.

The numerical value of temperature rise used in Eq. (10) was found from observing the temperature gauges across the vent condenser (Fig. 21) over four months. It was found that the average temperature rise across the vent condenser was 19 ^{o}F. However, the average temperature for the data taken was 18.4 ^{o}F (Appendix D). This value was approximated to 19 ^{o}F in the calculations. This approximation was made because the temperature was visually taken from temperature gauges installed across the vent condenser. Therefore, an error of ±1 ^{o}F was taken into consideration. See Appendix D for the vent condenser heat savings calculations. An example of these calculations was made for the purpose of following the same procedure to calculate the annual energy savings from the vent condenser. From having the annual energy savings, LCCA was performed to predict the total LCC in order to compare the savings from the vent condenser with that from the Grundfos pumps running in level control mode (Case 2).

2.3.2.2 Pressure Drop Calculations in Pipelines

Calculation of the pressure drop was necessary for estimating the extra pump horsepower needed for the constant speed pump to lift the water to the vent condenser and drive the water back to the storage tanks. Figure 45 shows the pipes and fittings that had been installed to deliver condensate water to the vent condenser. Therefore, an approach was presented in order to calculate the pressure at different points. These pressures were required to overcome the pipes and fittings friction losses as well as the elevation differences. Two points, A and B, were considered and are labeled in Fig. 45. The pressure at point B was assumed to be 2 PSIG. 2 PSIG was used as the estimated pressure value needed for the condensate water to reach the storage tanks.

The pipes' roughness and friction factor (ε, f) were unknown. However, pipe roughness and friction factor can be estimated from knowing the flow rate and the pressure drop between any two points. Therefore, the calculations were based on the known pressure drop between the constant speed pump's discharge line (the point where the Omega pressure sensor was installed) to the point right before the control valve (the point where the Danfoss pressure sensor was located). Also the flow rate was known from the Siemens flow meter readings. See Figure 17 for pressure sensor locations and Section 1.5 for the information on both pressure sensors and the Siemens flow meter. This approach was used to determine the pipes' properties (ε, f). Then, the pressure drop between points A and B could be calculated using the determined pipes' properties. Note that f was assumed to be constant based on the assumption that Re was larger enough for the f curves to be flat (Fig. 3).

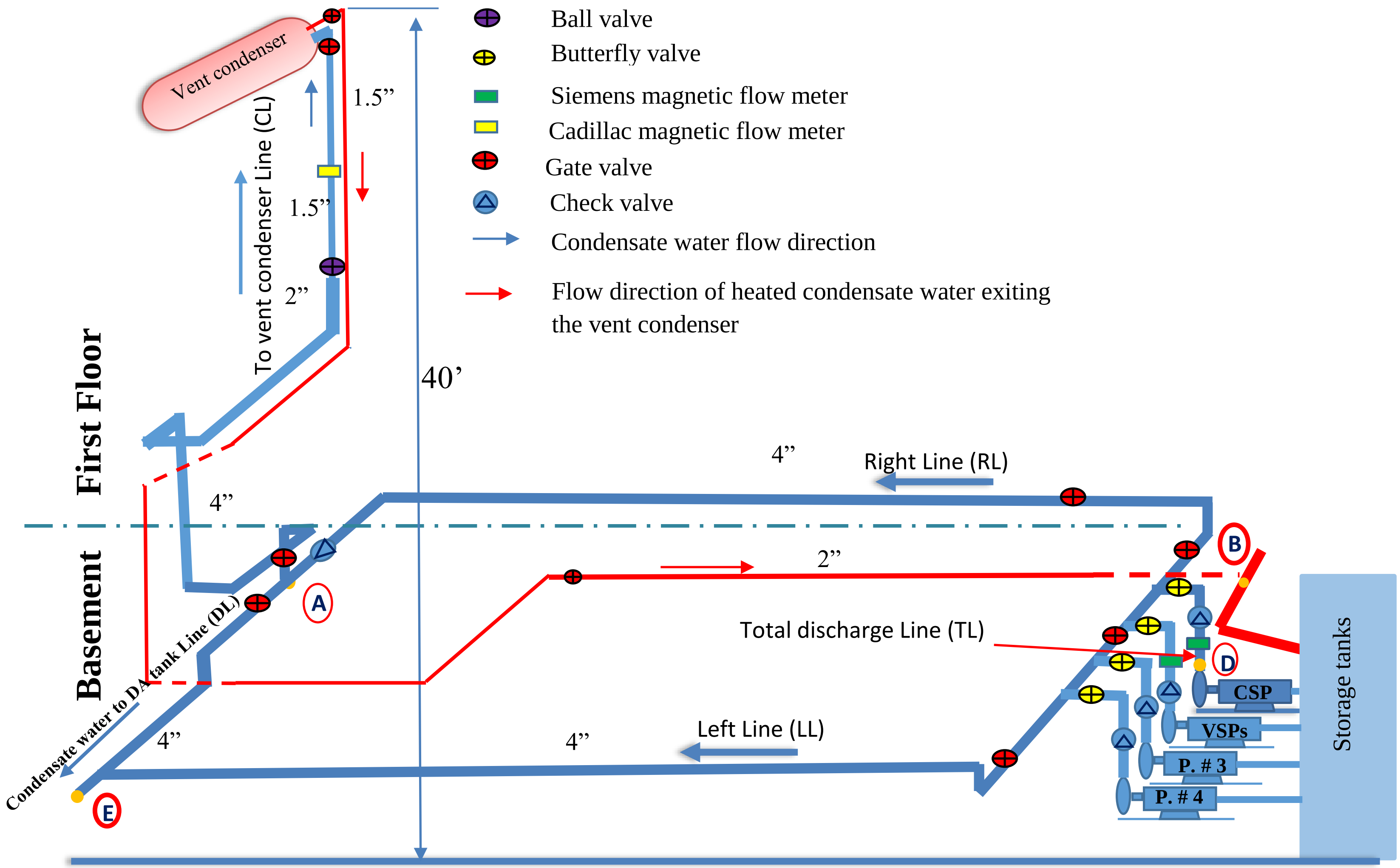


Figure 45: Vent condenser supply pipeline schematic

In order to calculate the pipes' properties, many assumptions were made. The first assumption was that the pipe connections were flanged connections instead of welded connections. The exceptions to this were the pipe lines that feed the vent condenser with condensate water and the line that carries the water back to the storage tanks. These pipes use threaded joints. The second assumption was that two of the gate valves (both installed just after the branch point at location A) were Milwaukee gate valves. This is because no information could be found on these valves, as they were too old. However, the other gate valves' information was considered and used for determining the local resistance coefficients (K).

The internal pipes diameters that were used are shown in Table 1 [56].

Table 1: Extra heavy pipe schedule 80-ASTM A53 type A [56]

NPS Designator*	DN Designator**	Outside Diameter		Inside Diameter		Wall Thickness	
		(inches)	(mm)	(inches)	(mm)	(inches)	(mm)
1-1/2"	40	1.900	48.3	1.500	38.1	0.200	5.08
2"	50	2.375	60.3	1.939	49.3	0.218	5.54
4"	100	4.500	114.3	3.826	97.2	0.337	8.56

*NPS stands for Nominal Pipe Size.

** DN stands for Diameter Nominal

The frictional head losses for different lengths of pipe where calculated using the Darcy-Weisbach formula of Eq. (3) [3].

The friction factor, f, in Eq. (3) is not constant and depends on the Reynold's number, Re. Almost all of the pipe flow conditions in the KU steam power plant were turbulent (Re> 4000 [3]). Therefore, f was found from the Moody diagram (see Fig. 3) or the corresponding equations for f vs. Re and ε (Eq. (13)). The calculations assumed that the flow was very turbulent. The pipes' friction factors were determined first, and then the Swamee Jain relationship, Eq. (13), was used to solve for the pipes' roughnesses [49].

$$f = \frac{0.25}{\left\{\log\left[\frac{\varepsilon}{3.7D} + \frac{5.74}{Re^{0.9}}\right]\right\}^2} \tag{13}$$

The selected discharge flow rate of 214 GPM in the line was not completely located in the very high Re zone. It was located in the transition zone. In the transition zone (see Fig. 3), f depends on both the Re and ε. Therefore, the calculations were carried out with acceptable errors. These errors resulted from assuming the friction factor to be constant for all pipe branches. In fact, the friction factor was not constant, it changed for each branch. This change was due to the fact that the flow rate was different in different pipe branches. See Appendix E1, E2, and E3 for comparison between the estimated f and the recalculated f depending on each pipe's flow rate and positon.

The pipe friction factor was back-calculated using Eq. (14) [49] from the known pressure head between the discharge point where the Omega pressure sensor was installed to the point where the Danfoss pressure sensor was installed.

$$\left(\frac{P}{\gamma}\right)_A - \left(\frac{P}{\gamma}\right)_B + H_{pump\ total} = (Z_B - Z_A) + \left(\frac{V_B^2}{2g} - \frac{V_A^2}{2g}\right) + \sum f\ \frac{L}{D}\ \frac{Q^2}{2gA^2} + \sum h_{minor} \tag{14}$$

All terms are defined in the Nomenclature. The subscripts A and B denote the locations A and B in pipeline (Fig. 45).

However, in order to calculate the minor losses, each fitting had to be addressed separately. Typically, minor losses are given in term of a local resistance coefficient, K, and the average velocity [3].

$$h_{minor} = K\ \frac{V^2}{2g} \tag{15}$$

In order to determine K for each fitting, Table 2 was made to show many of the equations that are used in calculating this coefficient.

Table 2: Example equations used to calculate the resistance coefficient, K, and the resulting values for different types of fittings

Fitting type	Fitting size	Cv $\left(gpm\sqrt{\frac{in^2}{lb}}\right)$	Equation used	K
Regular flanged 90° elbows	4”	N/A	N/A	0.3 [3]
Milwaukee Butterfly valve ML233 E	4”	860 [57]	$K = \frac{(29.9\frac{gpm}{\sqrt{lb}\,in})^2(DN)^4}{(C_v)^2}$ [3]	0.309
Milwaukee gate valve F-2885 M	4”	945 [58]	$K = \frac{(29.9\frac{gpm}{\sqrt{lb}\,in})^2(DN)^4}{(C_v)^2}$	0.256
Check valve F-2974-M horizontal swing -Iron	4”	605 [59]	$K = \frac{(29.9\frac{gpm}{\sqrt{lb}\,in})^2(DN)^4}{(C_v)^2}$	0.6257
Flanged T-section, Branch Flow	4”	N/A	N/A	1.0 [60]
Flanged T-section, Line Flow	4”	N/A	N/A	0.2 [60]
Screwed gate valve	2”	N/A	N/A	0.175 [3]
NIBCO gate valve F-637	2”	215	$K = \frac{(29.9\frac{gpm}{\sqrt{lb}\,in})^2(DN)^4}{(C_v)^2}$	1.063
90° threaded elbow	2”	N/A	N/A	0.9 [3]
Threaded 45° elbow	2”	N/A	N/A	0.3 [3]
Contraction 2”x1.5”	N/A	N/A	$K = (\frac{1}{C_c} - 1)^2$	0.176 [61]
Threaded 90° elbow	1.5”	N/A	N/A	1.1 [3]
Threaded 45° elbow	1.5”	N/A	N/A	0.3 [3]
Expansion 1.5”x2”	N/A	N/A	$K = (1 - (\frac{d'}{DN})^2)^2$ [3]	0.191 [3]

C_c is the contraction coefficient (from tables in Ref. 61). For the ratio of d'/DN =1.5”/2”, C_c is equal to 0.7045 [61].

All of the Table 2 equations were programmed in Excel so that the friction factor could be determined. If condensate water was fed to the vent condenser, the K for the T-section was taken to be that for branched flow. However, when the vent condenser was isolated (shut off) from the pumping system lines, the flow at location A in Fig. 45 changed from branch flow to line flow. The pipes' elevations and lengths were measured in the steam power plant and input into the Excel sheet (see Appendix E for pipes' length and elevation with respect to pump center lines). The flow rates in the pipes were taken from the data gathered in this project with an assumption that the flow in the total discharge line (TL) was divided equally between the left line (LL) and right line (RL) (Fig. 45). This assumption was acceptable since the left and right lines had the same length and elevation. Therefore, assuming half of the total flow rate in each line was reasonable.

After calculating all terms used in Eq. (14) except for friction factor, the friction factors for all pipes could be found using the 214 GPM flow rate in the total discharge line, and the recorded average pressures from the sensors were 36.56 PSIG and 30.33 PSIG for the discharge point (labeled D on Fig. 45) and the point right before the control valve (labeled E on Fig. 45). These pressure values were measured by the Omega and Danfoss pressure transducers at the 214 GPM flow rate. The friction factor for all pipes was 0.02198 (see Appendix E1). Then, Equation (13) was used to determine the pipe roughness (ε) for each branch. Pipe roughness was calculated depending upon the pipes' positions, pipes' diameters and flow values. In other words, pipe roughness was not considered to be constant for all of the 4" pipes, but each pipe line was considered to have a different pipe roughness depending on the pipe's position, the flow rate in that pipe line, and the pipe's diameter (Table 3).

Table 3: Pipes roughness

Pipe nominal diameter size, D (in)	Inside diameter size, D (in)	Pipe roughness, ε (in)
4" (TL)	3.826	0.00519
4" (LL)	3.826	0.00450
4" (RL)	3.826	0.00450
4" (DL)	3.826	0.00506
4" (CL)	3.826	0.00445
2" (CL)	1.939	0.00145
1.5" (CL)	1.500	0.00145

See Fig. 45 for pipelines designated by the symbols TL, LL, DL, RL, and CL that are used in Table 3.

After having all of the pipe properties, the Swamee Jain relationship, Eq. (13), was used to calculate the friction factor for any flow rate in each individual pipe.

Table 4 presents a comparison between measured and calculated (from Eq. (14)) pressures before the control valve. As discussed earlier, the pipe roughness had been determined. Therefore, in order to check the results shown in Table 3, six tests were performed. In each test, a different flow rate was used in order to check the calculated pressure drop between the same two points (discharge and before control valve) that were used to determine the pipes roughness. From these two sets of information, there was good agreement between the calculated data and the measured data, with their percentage errors being less than 2.0% as shown in Table 4. The percentage error was calculated based on the measured pressure values from the pressure sensors.

From Table 4, one could conclude that the procedure followed to calculate the pressure right before the control valve was accurate enough to follow the same procedure to calculate the pressure drop in line A-B (Figure 45).

Again, the calculated pressure drop in line A-B helped to determine the extra pressure needed to lift the water to the vent condenser and then return that amount to the storage tanks for a specific flow rate. From this pressure drop, power consumed could be computed. This will be explained in Chapter 5. See Appendix E for pressure calculation Excel sheets.

Table 4: Comparison between the calculated pressures from Equations (14) and (15) and the measured pressure from the pressure sensors

Trial Run #	Data Recorded Date	Total Discharge (GPM)	Vent Condenser Flow Rate (GPM)	Discharge Pressure (PSIG)	Measured Pressures	Calculated Pressures	Error %
					Pres. before Control Valve (PSIG)	Pres. before Control Valve (PSIG)	
1.	02/05/15	214.700	82.140	36.560	30.233	30.318	-0.280
2.	03/03/15	186.739	86.451	39.843	34.336	34.068	0.779
3.	02/08/15	167.355	89.235	41.910	36.831	36.408	1.147
4.	01/27/15	160.810	90.000	42.620	37.811	37.201	1.614
5.	01/27/15	107.693	0.000	46.682	41.682	41.568	0.273
6.	03/03/15	73.260	0.000	48.390	43.931	43.562	0.840

All results in Table 4 are taken from Appendix E2.

The pressure drop between points A-B was then calculated. However, there was no pressure sensor in the line going to the vent condenser (from location A to the vent condenser) to check the obtained results. The first assumption made in these calculations was that the pressure drop across the vent condenser was 2 PSIG; and the pressure at point B was assumed to be approximately 2 PSIG. In order to have the flow in the recirculation line [the vent condenser return line] reach the storage tank, calculations must show that the pressure at the intersection point where the recirculation line meets the condensate return was equal to 2 PSIG. Otherwise the condensate water in the return line would flow into the vent condenser line (Fig. 45). Using the data recorded from the Worthington pump, its discharge pressure was 36.56 PSIG for a flow rate of 214 GPM and a vent condenser flow rate of 82.1 GPM. Calculating the pressure drop between points A and B, it was found that the minimum pressure at point A that must be produced by the Worthington pump in order to lift condensate water to the vent condenser was 28 PSIG. This calculation was by trial and error in order to have the pressure at point B roughly equal to 2 PSIG. That means the pressure drop between A and B was roughly 26 PSIG. (See the Excel sheet calculations in Appendix E3 for details.) The equations used in calculating the pressure drop are not presented in Appendix E. The graduate research of Allabdullah present these equations in more detail [62].

In summary, for a flow rate of 82.1 GPM to the vent condenser, the pressure drop was found to be 26 PSIG. In order to increase the flow rate to the vent condenser, the pressure at point A must increase. The calculation procedure of Appendix E3 can predict the pressure drop for any flow rate based upon the assumption that pressure drop across the vent condenser was 2 PSIG and did not change when the flow rate changed.

2.4 Life Cycle Cost Analysis Procedure and Equations

In order to determine the most cost effective pumping system and control method that should be used in the KU steam power plant, LCC was employed. In this project, LCCA was performed using the BLCC5 program provided by the National Institute of Standards and Technology (NIST), which can be downloaded from Ref. 36 free of charge. This economics software was developed in order to help in making decisions in selecting alternatives, balancing initial capital costs against operating and maintenance costs over a specific life time [36].

This section provides the related costs involved in determining LCC. The equations used in the BLLC5 program to calculate the present value from future costs that will be detailed in this section for a time period of twenty years [24]. For projects that are directed toward evaluating energy savings, the FEMP (Federal Energy Management Program) Analysis Energy Project of the BLCC5 is applicable. FEMP is more involved in evaluating a life cycle costs for energy and water conservation and renewable energy projects similar to the project at hand [29].

It is crucial to realize that the study period has two important dates: the base date and the service date. The base date is the date from which "all project-related costs are discounted in LCCA" [29]. Obviously, it is important to select the same base date for the project alternatives that are under study. The other date is the service date. This date is similar in concept to the base date and sometimes these dates are selected to be the same, as shown in Fig. 46 [29]. The service date can be defined as "the date on which the project is expected to be implemented. Operating and maintenance costs (including energy and water related cost) are generally incurred after this date not before." [29]. Before including all relevant costs in LCCA in Eq. (6) [29], these costs must be discounted to their present value because they are occurring at different times in the future.

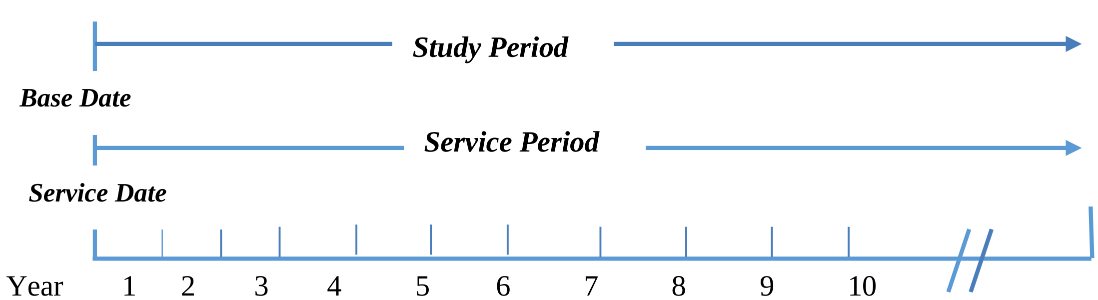


Figure 46: Coinciding study period and service period (reproduced from Ref. 29)

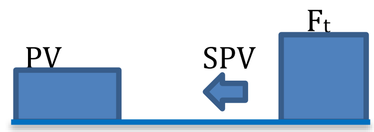
In order to discount the future values into their present value equivalents, BLLC5 uses four discounting factors as shown in Table 5 [29]. These factors are: 1) Single Present Value (SPV) factor, 2) Uniform Present Value (UPV) factor, 3) Uniform Present Value factor modified for price escalation (UPV^*), and 4) FEMP UPV^* factor for use with energy costs [29]. Each factor is used for a different type of amount, depending upon the occurrence frequency in the future, and is explained in Table 5 [29].

On the left side of Table 5, there are formulas that can be used for determining each type of factor. Tables Ba-1 to Ba-5 of Ref. 29 are updated annually and uploaded by DOE. They can be found in Ref. 36 under the “Annual Supplement to Handbook 135”. For this project, these tables were updated in 2014 [63]. BLCC5 users do not have to calculate these factors manually because DOE enters all of them in the program. Moreover, these factors are computed for a study period range (1- 30 years) using the latest FEMP discount factors and energy price escalation rates. Therefore, it is important to download the latest updated program to perform the most accurate LCCA.

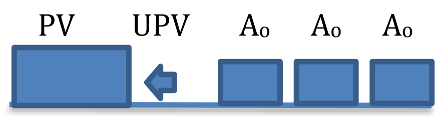
The first cost to be input in to the LCCA program was the initial investment occurring one only time. There was no need to discount this cost to present value as it was already a present value. The initial investment costs of the constant speed pump were as follows. The Worthington pump cost was US$2,500 plus the cost of the control valve or flow regulator valve that had to be employed with it in order to control/regulate the flow to the DA tank according to the demand. The cost of the Fisher control valve was US$4,000 [10]. The other initial cost that is associated with the constant speed pump is the control valve installation cost. Part of the installation cost was a certificated welder to have the control valve flanges welded. The control valve installation cost, including the cost of two flanges, was estimated to be US$6,000 [30].

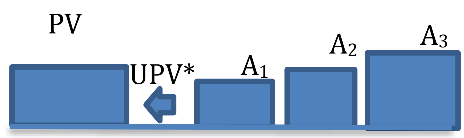
Table 5: Present-value formulas and discount factors for LCCA (reproduced from Ref. 29)

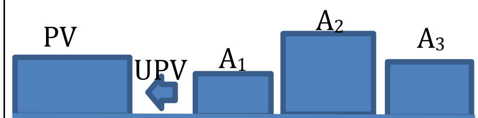
<table>
<tr>
<td>PV formula for one-time amounts

The Single Present Value (SPV) factor is used to calculate the present value, of a future cash amount occurring at the end of the year t. F_t, given a discount rate, d.
$$PV = F_t \frac{1}{(1+d)^t}$$</td>
<td>$$PV = (F_t)\ SPV_{(t,d)}$$


The SPV factor for d=3% and t= 15 years is **0.642**.</td>
</tr>
<tr>
<td>PV formula for annually recurring uniform amounts

The **Uniform Present Value (UPV) factor** is used to calculate the PV of a series of equal cash amounts, A_o, that recur annually over a period of n years, given d.
$$PV = A_o \sum_{t=1}^{ni} \frac{1}{(1+d)^t} = A_o \frac{(1+d)^{ni}-1}{d(1+d)^{ni}}$$</td>
<td>$$PV = A_o UPV_{(ni,d)}$$


The **UPV factor** for d=3% and ni= 15 years is **11.94.**</td>
</tr>
<tr>
<td>PV formula for annually recurring **non**-uniform amounts
$$PV = A_0 \sum_{t=1}^{ni} (\frac{1+e}{1+d})^t = A_0 \frac{(1+e)}{(d-e)} [1 - (\frac{1+e}{1+d})^{ni}]$$
The **Modified Uniform Present Value (UPV*) factor** is used to calculate the PV recurring annual amount that change from year to year at a **constant** escalation rate, e (i.e., $A_{t+1} = A_t\ (1+e)$), over n years, given d. the escalation rate can be positive or negative.</td>
<td>$$PV = (A_o)\ UPV^*{}_{(ni,d,e)}$$


The **UPV* factor** for e= 2%, d=3%, and ni= 15 years is **13.89.**</td>
</tr>
<tr>
<td>PV formula for annually recurring energy costs (FEMP LCCA)
The **FEMP UPV* factor** is used to calculate the PV of annually recurring energy costs over n years, which are assumed to change from year to year at a non-constant escalation rate , based on DOE projects. FEMP UPV* factors for the current DOE discount rate and published in Tables Ba-1 through Ba-5 of the Annual supplement to Handbook 135 [36].</td>
<td>$$PV = (A_o)\ UPV^*{}_{(reg,ft,rt,d,ni)}$$


The **FEMP UPV* factor** for region (reg)=3, fuel type (ft)= electricity, rate type (rt)= commercial, d= 3%, and ni= 15 is **12.12** (1995).</td>
</tr>
</table>

The two Grundfos variable speed pumps and the central panel with CR monitoring controller were US$15,000 [10]. Both pumps installation costs were assumed in this thesis to be $1,000. In order to control the condensate water level in the DA tank, a level sensor was needed for the Grundfos pumps. However, the control valve that was needed for the Worthington pump also required a water level sensor to signal the control valve in order to regulate its operation. Thus, a level sensor was assumed to be necessary for both pumps' operation. For this reason, the cost of the level sensor $1,972 [64] was added to each type of pump's initial cost (the level sensor invoice can be found in Appendix K). The residual value in Eq. (6), Res, was not considered in this comparison, assuming there would be no residual value at the end of the twenty years study period [10].

The operation, maintenance and repair (OM&R), and replacement costs (Repl) in Eq. (6) were considered. For example, replacing a pump mechanical seal costs US$200 for either pump. The replacement of motor or impeller seals routinely occurs every two years; and the labor to carry out the replacement was estimated to be US$1000 for the Worthington Pump and US$2000 for the Grundfos pumps. However, for this project, the replacement was averaged over every ten years, starting from the date of the pumps' installation. The seal replacement labor was considered as "non-annually recurring costs" [29], i.e., costs that were not occurring annually but occurring at irregular times [29]. On the other hand, the costs of labor (US$1,000) for both pumps every year was considered as an "annually recurring cost" [10].

Each of these costs in the previous discussion were treated differently according to the frequency of occurrence. The equations described in Table 5 used "DOE's real discount rates (excluding general price inflation), nominal discount rate (including general price inflation) and implied long-term average rate of inflation for 2014 of 3.0%, 3.1% and 0.1%, respectively" [63].

The last set of costs are the energy and water consumption for both pump systems, E and W, in Eq. (6). The costs were obtained by contacting the City of Lawrence water supply division. These costs were US$0.0736 per kWh [65], US$480.00 Annual Demand fees [10], and $3.39/1,000 gallons of water [66].

The annual energy consumption was calculated based on the power consumption data gathered in the project. Power consumption data was gathered during the time at which the steam demand was at its highest, i.e., winter time. In order to determine the average energy consumption for the months in which the power consumption data was not gathered, an estimate was made based on the power consumption data gathered for the months that had the highest energy consumption and the total steam generated in all twelve months (to be explained in Section 4.1). This approximate calculation was employed in order to estimate the average annual energy consumption. Tables of steam generated by the steam power plant are presented in Appendix F.

The LCC for the vent condenser was calculated separately in order to have an estimate as to how much annual savings the vent condenser was providing, and then compare that savings with the savings that the steam power plant might have when running the variable speed pumps in level control mode in Case 2 without the vent condenser. The results from this comparison would show whether the savings from running the variable speed pumps was greater or less than the savings from the energy reclamation resulting from using the vent condenser.

In order to find the total LCC of the vent condenser, assumptions were made. The total LCC time period of the vent condenser was assumed to be 10 years instead of 20 years. Therefore, in order to have the same LCC time period for both pump systems and the vent condenser, a second vent condenser was assumed to be purchased at the end of first 10 years. The initial cost of the vent condenser with all piping and fittings requirement along with the labor work was assumed to be $10,000 [22]. Another $10000 was assumed to purchase and install. The second vent condenser at 10 years from the start considered price increases due to inflation. Because there would be a little maintenance required for the vent condenser, an average of $2500 [22] in maintenance fees was assumed for every two years to check the vent condenser tube bundle, and replace the rusted tubes if necessary. Again, no residual value was assumed for the vent condenser [22]. The maintenance costs were considered as “non-annually recurring costs” [29]. The natural gas cost was obtained from the energy engineer of the University of Kansas, which was 0.53 cent/ft^3 [57].

The LCCA of the vent condenser is different in concept than that of the pumps because the operating costs that are associated with running the pumps are costs that the investor has to pay. On the other hand, the operating “costs” of the vent condenser are not paid by the investor, but are saved due to purchasing less natural gas than if there were no vent condenser (Section 2.3).

In order to calculate the savings from running the vent condenser, the data gathered from the Cadillac magnetic flow meter that measures the condensate water flow rate was used to calculate the mass flow rate of water to the vent condenser (see Appendix D example). For the months in which the Cadillac magnetic flow meter was not installed (May, June), an estimate was made to consider the condensate water flow rate to be approximately the same flow rate as for the nearest month in which the data was available. For instance, the flow rate in May of 2014 was approximated to be the same as the flow rate in June of 2014.

The Cadillac magnetic flow meter was installed in the steam power plant in July of 2014. Therefore, the data was available in July through September then from November through April. As discussed in Section 2.2, expanding the testing system and calibrating the Siemens flow meter was performed in October of 2014. Therefore, the work was concentrating on the calibration, and no data was gathered during this month. However, in order to estimate the condensate water flow rate in October, the data gathered in September and November were averaged. The other factor that was used to calculate the annual energy gain from the vent condenser was the boiler efficiency that helped to determine the amount of natural gas that would have been consumed in order to generate the corresponding amount of energy in the steam. The boiler efficiency for each month was calculated in Appendix F for each day. Then the monthly average boiler efficiency was calculated from averaging the daily boiler efficiencies. The monthly boiler efficiency range was from 88 % to 90%. See Section 2.3.2.1 for the equations used to calculate the boiler efficiency, and Appendix D for the vent condenser's energy reclamation calculation.

The Base Date for the study period was April of 2014 and the study period was 20 years for both the pumps' LCC and the vent condenser's LCC. The constant dollar method was used to calculate the LCCs as it is "supported by the BLCC computer program" [29]. The constant dollar method requires no estimate of the inflation rate. The future costs in this analysis are discounted using "real discount rate that excludes the rate of inflation" [29].

Chapter 3: Results

3.1 Case 1

3.1.1 Test #1, February 5 to February 14, 2015

Case one presents a comparison between the Worthington constant speed pump's and the Grundfos variable speed pumps' power consumption, total flow rate, discharge pressure, and vent condenser flow rates. Both pumps were running in the pressure control mode, with the vent condenser valve open, and the DA control valve regulating the flow. The data was gathered from February 5 to February 14 of 2015.

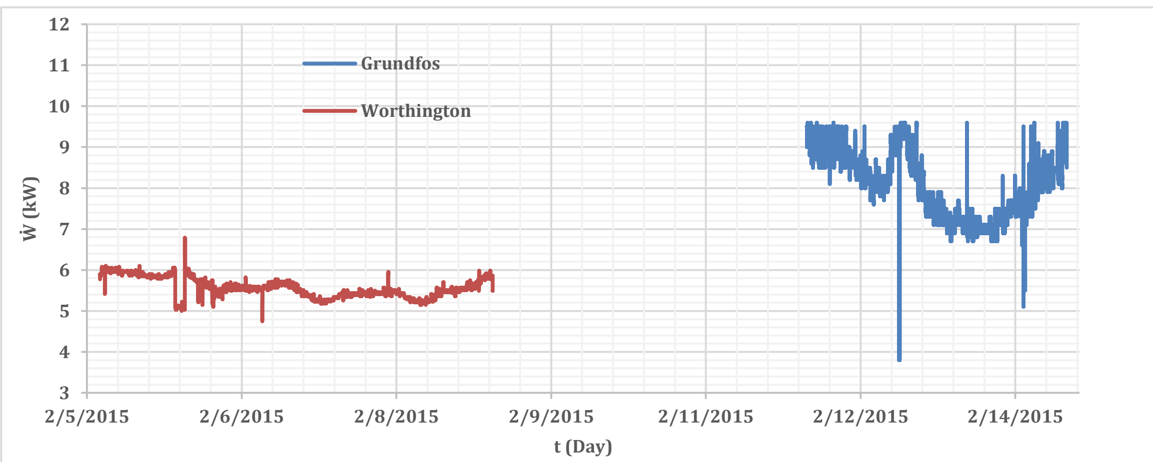


Figure 47: Power consumption for Grundfos and Worthington pumps from Feb. 5 through Feb. 14 of 2015 (Case 1, Test #1)

As evidenced from Fig. 47, the Worthington constant speed pump was running from February 5 through February 9. Then the Grundfos pumps took over the work from February 12 through February 14. Because the steam power plant was running two pumps at the time when the steam demand was high, the power consumption data for the Grundfos pumps was limited only to two days (Fig. 47). The remaining days of the week in which the Grundfos pumps were running are not presented in Fig. 47 because there was a second pump running with the Grundfos pumps.

The average power consumption of the Worthington pump was 5.567 kW while the average power consumption of the Grundfos pumps was 8.433 kW. The Grundfos pumps consumed more power than the Worthington constant speed pump because the generated steam was higher in the time when the Grundfos pumps were running. This can be clearly seen from Fig. 48. The Worthington pump flow rate was not completely constant. The Grundfos pumps were providing a high flow rate to meet the demand and the desired set point pressure of 43 PSIG.

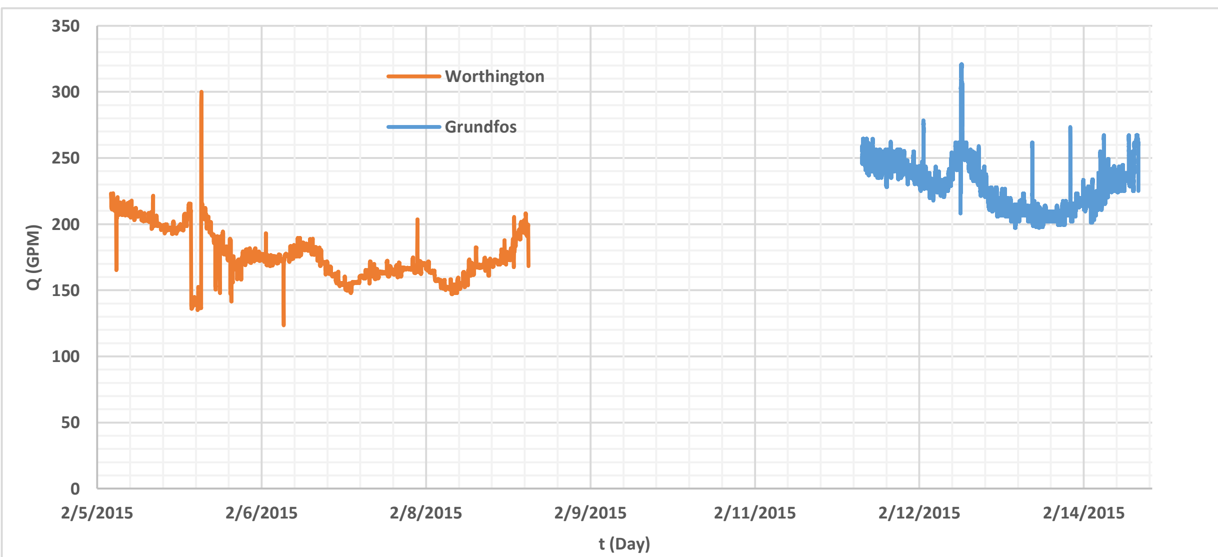


Figure 48: Flow rates for Grundfos and Worthington pumps from Feb. 5 through Feb. 14 of 2015 (Case 1, Test #1)

The average flow rate of the Worthington constant speed pump was 176.225 GPM while the average flow rate of the Grundfos pumps was 228.759 GPM. These two different values justified the higher power consumption of the Grundfos pumps. Even with this explanation, in later sections, more analysis will be performed to compare the actual power consumption measured when the pumps were running with data shown in pumps' characteristic curve (from the manufacturers). Moreover, the pumps' hydraulic power will also considered in order to determine the theoretical pumps' power consumption and compare with the recorded power consumption.

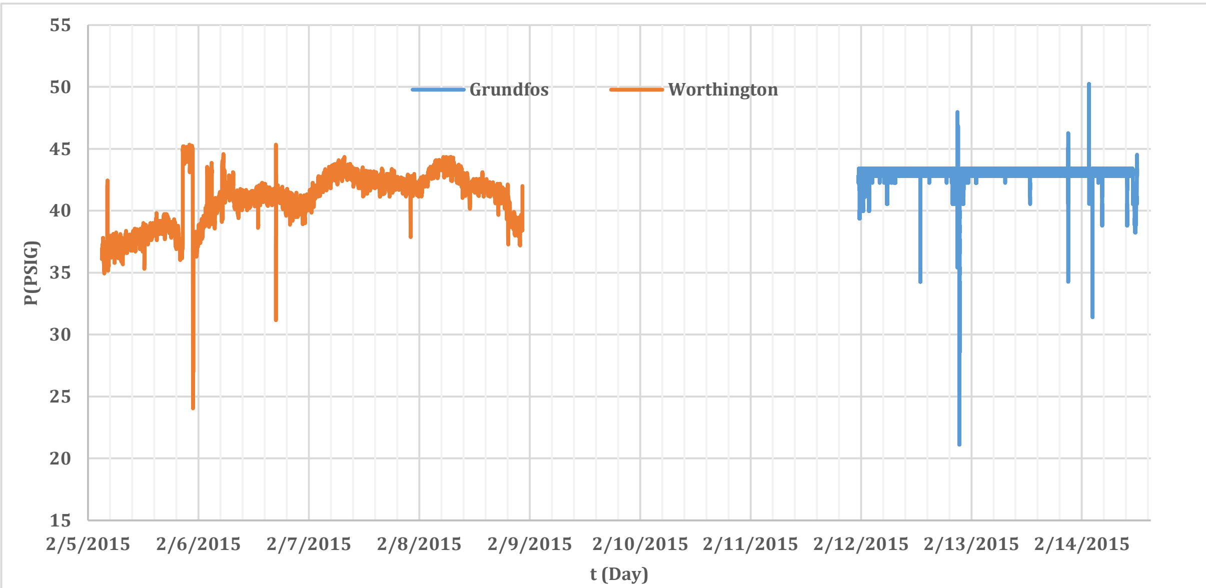


Figure 49: Discharge pressure for Grundfos and Worthington pumps from Feb. 5 through Feb. 14 of 2015 (Case 1, Test #1)

Figure 49 presents the discharge pressure of the Worthington pump which varied often based upon the pump's operating point, and changed depending upon the system curve changes. The operating point changed due to the control valve. The control valve changed its opening percentage to keep the water level in the DA tank fairly constant (~ 52%). When the control valve opening increased, that caused the system curve to become flatter; and when the control valve opening decreased, the system curve became steeper. The system curve trend changed due to the frictional losses generated by the control valve. It is important to note that the control valve opening positions changed due to water level in the DA tank changing with a variation that corresponded to the boilers' demand.

On the other hand, the Grundfos pumps attempted to run at constant discharge pressure because of the operating mode selected by the steam power plant's staff. For example, when the steam demand increased, the variable speed pumps responding to that sudden increase and provided a higher flow rate at the same discharge pressure of 43 PSIG. That operating mode caused the Grundfos pumps to speed up to meet the high demand at a relatively high discharge pressure. On the other hand, the Worthington pump did not undergo such changes in operating conditions. If the steam demand increased, the constant speed pump would try to deliver a high flow rate; but the discharge pressure reduced due to the fact that this pump could not perform more work than that for which it was designed. See Fig. 48 and Fig. 49 on February 5 when the average flow rate was 207.31 GPM, and the average discharge pressure was 37.42 GPM for the Worthington pump. On the other hand, the average flow rate for the Grundfos pumps was 245.285 GPM on February 12, and the average discharge pressure was 42.913 PSIG. This operation condition makes the Grundfos pumps perform more work than the Worthington pump. As a result, the Grundfos pumps consumed more power. In addition, the comparison was not made for the same time period. For these reasons, this case study was excluded from the LCC in Chapter 4.

The overall average discharge pressure for the Worthington pump was 40.943 PSIG from February 5 through February 9, and the average discharge pressure of the Grundfos pumps was 42.7158 PSIG from February 12 through February 14, 2015.

Figure 50 presents the differential pressure drop across the control valve when both pumps were running, for the discussed period of time. The average differential pressure across the valve that controls the flow to DA tank was 26.685 PSIG and 27.411 PSIG for the Worthington pump and Grundfos pumps, respectively.

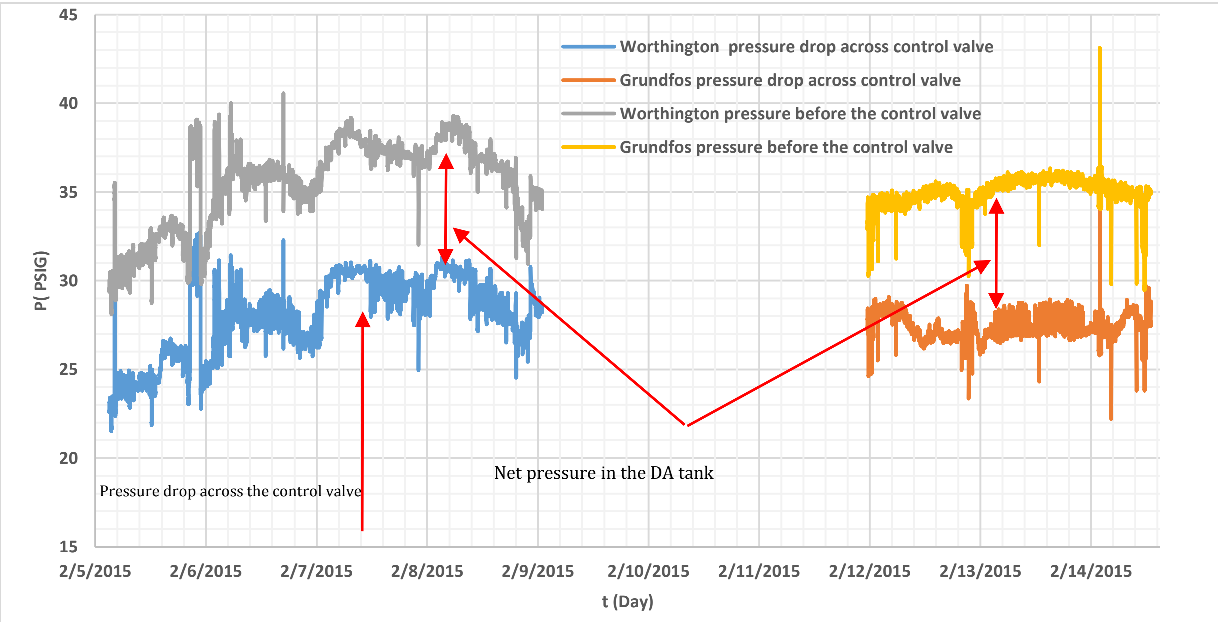


Figure 50: Differential pressure across the control valve for Grundfos and Worthington pumps from Feb. 5 through Feb. 14 of 2015 (Case 1, Test #1)

As evidenced from Fig. 50, the pressure drop across the control valve was fairly constant when running the Grundfos pumps, and slightly fluctuating when running the Worthington pump. This is because of the fact that the Grundfos pumps always delivered the same pressure to the control valve; and the control valve was trying to maintain the same pressure drop in order to regulate the flow into the DA tank. In other words, when the steam demand increased, the Grundfos pumps were attempting to increase the flow rate to meet the demand; but the discharge pressure at the pumps' outlets remained constant regardless of the change in flow rate. As a result, the control valve maintained the same pressure drop. On the other hand, when the Worthington pump was running, the pressure at the discharge was not constant and changed depending upon the steam demand or the control valve opening percentage. Therefore, when the steam demand increased, the control valve opened more trying to feed the DA tank with more condensate water. That increase in flow rate caused the constant speed pump's discharge pressure to decrease (as explained earlier). As a result, the pressure before the control valve consequently dropped. In this case, the control valve was trying to keep the pressure in the DA tank almost the same. The DA tank's pressure was selected to be 7-8 PSIG by the steam power plant staff. One should notice that the flow rate across the control valve varied, but the exit pressure of that flow remained fairly constant on the outlet side of the valve.

The pressure drop across the control valve and the pressure before the control valve followed the same pattern. Therefore, in the following results, the pressure drop across the control valve will not be presented unless there were some changes in the pressure drop that need to be discussed. The information just presented was designed to show how the pressure drop across the control valve might change, and how the control valve worked.

Next, Fig. 51 is an example of the condensate water flow rate to the vent condenser when either the Worthington pump or the Grundfos pumps were running with the control valve on duty.

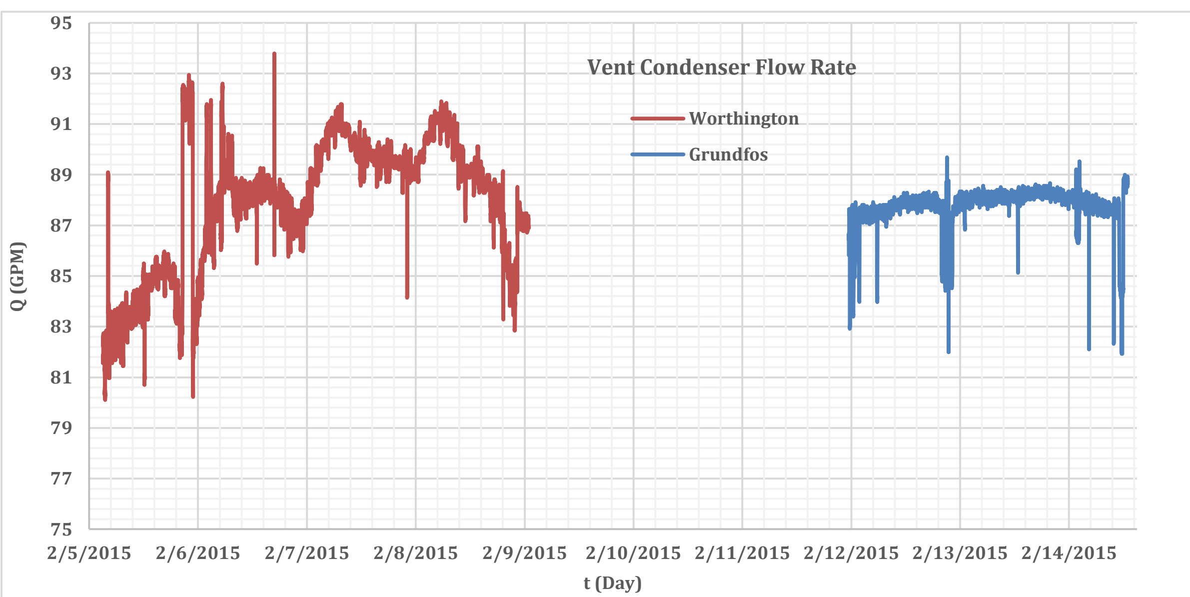


Figure 51: Vent condenser flow rate for Grundfos and Worthington pumps from Feb. 5 through Feb. 14 of 2015 (Case 1, Test #1)

The average vent condenser flow rate when the Grundfos pumps were running was 87.769 GPM while, when the Worthington pump was running, the average flow rate was 86.332 GPM. The overall average flow rate to the vent condenser was almost the same for both pumps. That is true because the amount of water that reached the vent condenser was directly related to the overall average pump discharge pressure. Therefore, because both types of pumps had nearly the same overall average discharge pressure, their abilities to lift water to the vent condenser were the same.

Again, the information gathered from the vent condenser followed the same pattern as shown in Fig. 51. Therefore, in the following results given for Case 1, these data will not be presented unless necessary.

Figure 52 presents the Grundfos pumps' inlet pressure, which will be taken as a constant value for this entire project. This pressure is the same for both types of pumps, even though the pressure sensor was installed only at the inlet side of the Grundfos pumps. The data was taken during December of 2014. Also, these data was taken at other times with the same results.

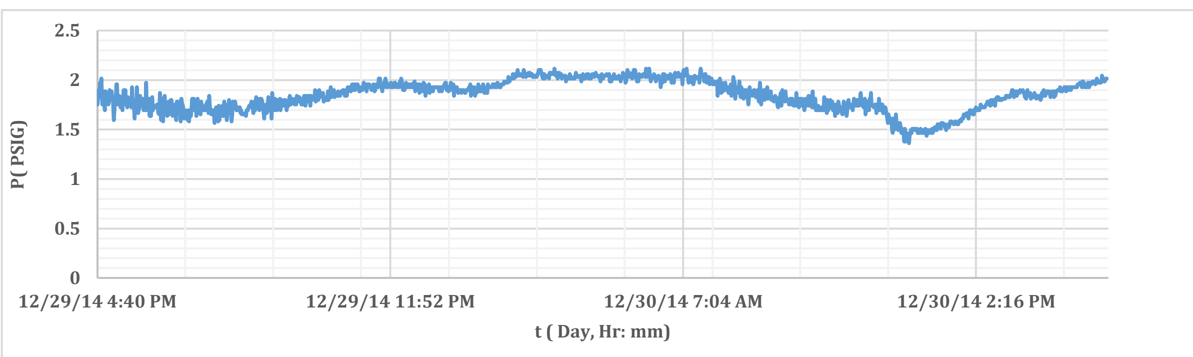


Figure 52: Inlet pressure for Grundfos pumps from Dec. 29 to Dec. 30 of 2014

As it is evident from Fig. 52, the inlet pressure is fairly constant and does not change significantly with time. The average inlet pressure for the Grundfos pumps was 1.846 PSIG from December 29 through December 30 of 2014. Therefore, the inlet pressure will be considered constant for both types of pumps in this project wherever needed in calculating pump power.

In order to check the validity of the recorded power consumption data, the system (pump and motor) power consumption was calculated using two approaches. The first was by reading the power consumption from pump manufacturers' performance curves (see Appendix A1 and A2). The second approach was using pump pressure and flow rate in a corresponding equation (to be explained in this section).

For the first approach, in order to be able to specify the average operating point, one must have two pieces of information, the average discharge pump pressure and the flow rate. The average discharge pressure is shown on the pump curves as pressure head in feet. Therefore, the average recorded discharge pressure given in PSIG was converted to feet. The term head can be defined as "the quantity used to express the energy content of the liquid per unit weight of the liquid referred to any arbitrary datum" [3]. To convert the pressure in pounds per square inch to feet, use [3].

$$H\ (ft) = P\left(\frac{lb}{in^2}\right)\frac{2.31(\frac{in^2}{lb}ft)}{\gamma'} \tag{16}$$

All terms are defined in the Nomenclature. The specific gravity (γ') for water at 160^0 F [10] is 0.979 [3]. After converting the discharge pressure to feet of head, and having the average flow rate at that pressure, one can use pump performance curves in Appendix A (A1 and A2) in order to find the required power to perform the work.

The second approach for validating the recorded power consumption uses flow rate and pressure to calculate the brake horsepower for a pump Eq. (17) [3].

$$bhp = \frac{Q\ (GPM)\ [\Delta P(PSIG)]}{C_1\ \eta_p} \tag{17}$$

C_1 is a units conversion factor in Eq. (17), and it is equal to 1714 ($\frac{GPM\ lb}{in^2\ hp}$).

However, in order to calculate the actual electrical power consumption by the system (pump and motor), the motor, pump, and VFD efficiencies are required. Therefore, Eq. (17) becomes [16]

$$\dot{W}(HP) = \frac{Q\ (GPM)\ [\Delta P(PSIG)]}{C_1\ \ (\eta_{m)}(\eta_V)(\eta_p)} \tag{18}$$

All terms are defined in the Nomenclature. Subscripts *p, m,* and *V* denote pump, motor and VFD, respectively.

According to the Grundfos pump performance curves, the overall efficiency of the system including the pumps, motor and VFD is given as “Eff. Pump & mtr”. Therefore, there is no need to calculate each efficiency individually. Using the Grundfos pump curves that are available in Appendix A2, one can find the power consumption of the system by directly using the recorded average flow rate and discharge pressure.

For Test #1, in order to validate the recorded power consumption for the Grundfos pumps, the pump performance curves show that the pumps’ electrical power consumption was 7.78 kW (10.433 HP). However, the obtained power value from the curves differs from the recorded power consumption of 8.433 kW (11.38 HP) being 7.74% lower. Using the same method to find the power consumption of the Worthington pump in Test #1, the power consumption from pump curves was approximately 5.011 kW (6.72 HP), while the average power consumption from the recorded data was 5.567 kW which is equivalent to 7.465 HP, showing that the recoded power was 9.98% higher than the power read from the pump curve. Don't forget the aging factor that also contributes to calculation errors because manufacturers do not give a factor that shows how pumps perform as they become old. The Grundfos pumps were installed in 2010, and the Worthington pump was installed in 2005.

In the second approach, Eq. (18) was used. Before using Eq. (18) to calculate the pump power consumption and compare to the other two power consumption values, the pump efficiency must be determined. The Worthington pump's performance curve shows that the mechanical pump efficiency at that operating point [found from the average flow rate and the discharge pressure from Test #1] is approximately 67% and the Worthington pump motor efficiency is 87.5% [67]. Therefore, after inputting the gathered average data for flow rate and discharge pressure for the Worthington pump and assuming that the inlet pressure to this pump during the time the data was gathered was 1.8 PSIG in Eq. (18), the Worthington pump power consumption was computed to be 6.864 HP (5.118 kW). Therefore, the calculated power consumption from Eq. (18) is about 8 % less than the recorded power consumption value of 5.567 kW for Test #1 and is 2% more than the power consumption obtained from the pump performance curves.

Following the same procedure with the Grundfos pumps for using Eq. (18), the combined pump and motor efficiency obtained from the Grundfos pump performance curves was 54.6%. The average flow rate and discharge pressure in Test #1 were 228.759 GPM and 42.913 PSIG, respectively. Equation (18) gives the power consumption of the Grundfos pumps as 10.05 HP which equivalent to 7.494 kW. This value is less than the recorded power consumption of 8.433 kW by 11.13% and is about 3.8% less than the power from the pump curve.

These types of calculations will be used in the results for all of Chapter 3 to ensure that the average power consumption of the recorded data is fairly accurate and comparable to both the information from the pump curves in Appendix A and the power consumption obtained from Eq. (18).

3.1.2 Test #2, March 3 through March 16, 2015

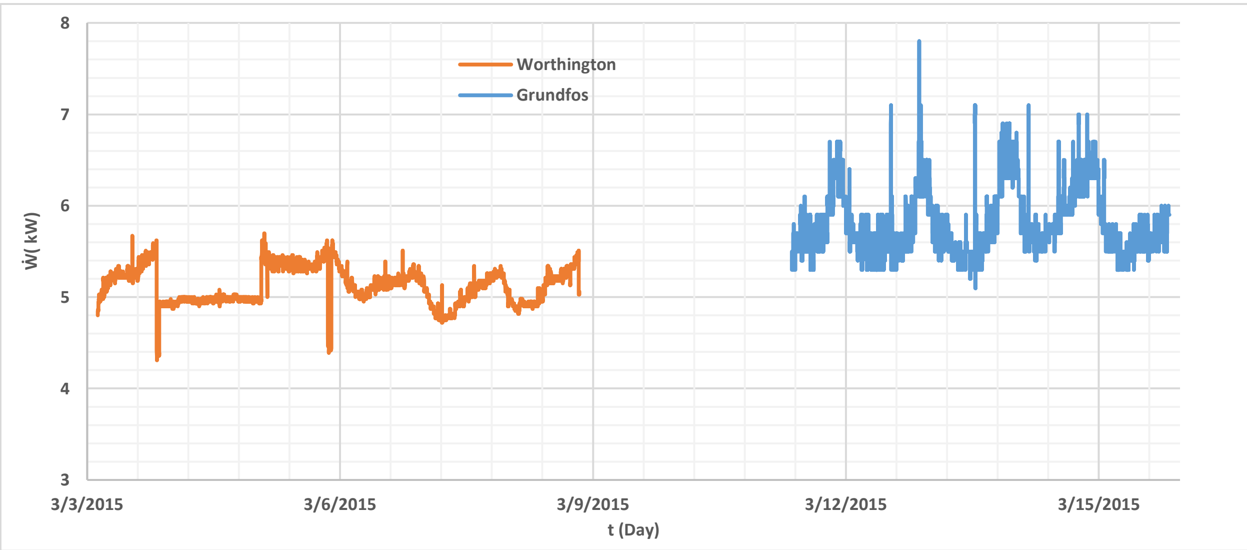


Figure 53: Grundfos and Worthington pumps' power consumption from March 3 through March 16, 2015 (Case 1, Test #2)

Figure 53 presents the power consumption of the two types of pumps, Grundfos and Worthington, as they were running during normal operation. For this period of time, the Worthington pump ran from March 3 through March 9. Then the Grundfos pumps took over the work from March 11 through March 16. The average power consumption of the Worthington pump was 5.121 kW, whereas the average power consumption for the Grundfos pumps was 5.796 kW. As is shown from the averages, the Grundfos pumps were consuming 11.6% more energy than the Worthington pump, even though the average flow rates were almost the same (Fig. 54). However, as explained previously, the Grundfos pumps could not run below the set point of 43 PSIG. On the other hand, the Worthington pump delivered a high flow rate to the DA tank while running at different discharge pressures (Fig. 55).

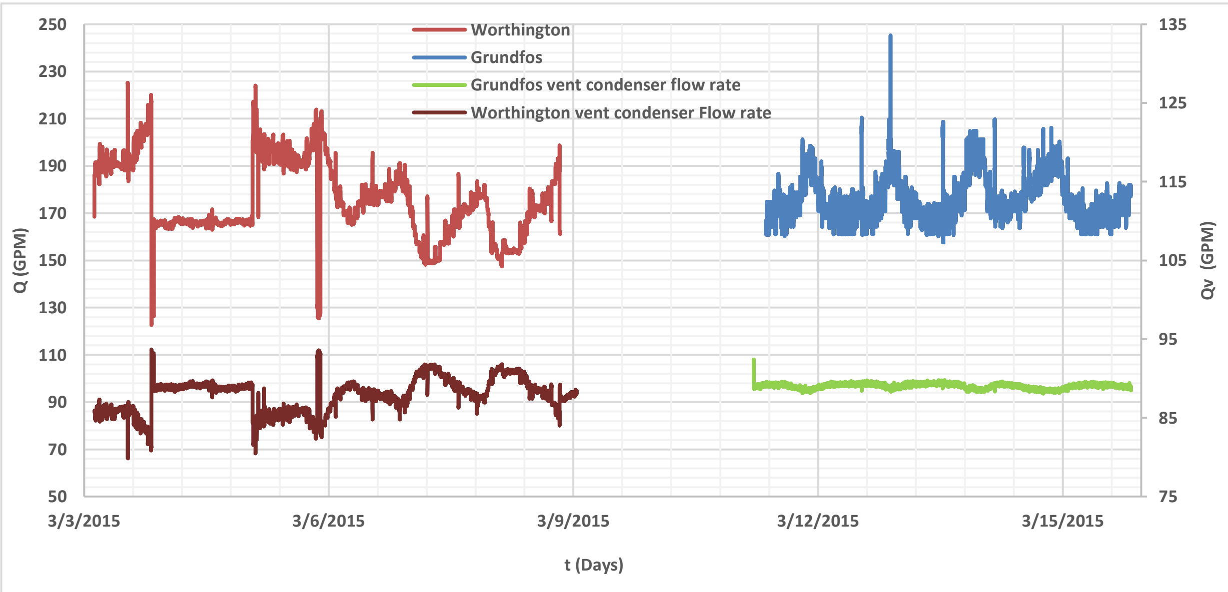


Figure 54: Grundfos pumps', Worthington pump's and vent condenser's flow rates from March 3 through March 16, 2015 (Case 1, Test #2)

The average flow rate of the Worthington pump for the specified time was 175.392 GPM; and the average flow rate for the Grundfos pumps was 175.878 GPM. These values show that the average flow rates for both pumps were almost the same. Figure 54 gives a clear picture of the peak steam demands that the steam power plant produced daily, especially in the Grundfos pumps' case. These peaks typically occurred every day in the morning when the students, faculty and staff arrived at the university and started opening the buildings' doors, causing the heating load to increase with a corresponding increase in the steam demand. Then steam demand dropped to a minimum when the students, faculty and the staff left the university, keeping the buildings' doors mostly closed. Also, in Fig. 54, the flow to the vent condenser is shown. One can see the amount of condensate water recirculating through the vent condenser. The advantage of this recirculation is that, during hot weather when the steam demand is at its minimum, instead of over pressurizing the discharge pipeline, most of the flow is recirculated through the vent condenser, helping to relieve the pressure in the pumps' discharge line.

The average flow rates to the vent condenser were 89.039 GPM and 87.9 GPM for the Grundfos and Worthington pumps, respectively, for the specified time period.

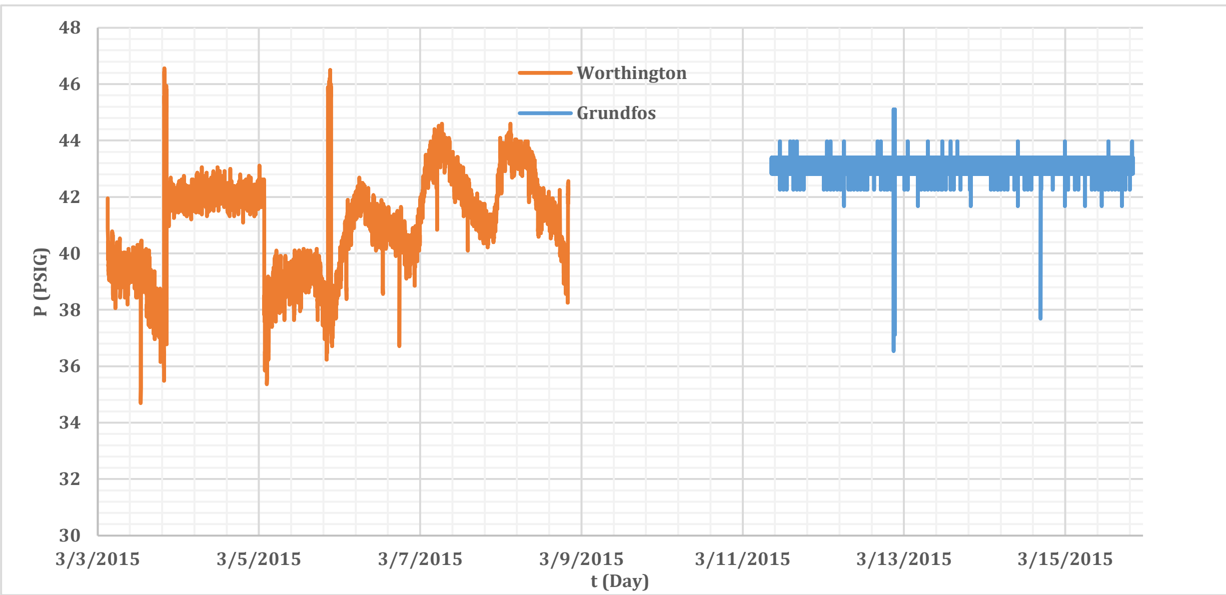


Figure 55: Grundfos and Worthington pumps' discharge pressures from March 3 through March 16, 2015 (Case 1, Test #2)

Figure 55 shows the Grundfos pumps' response as they attempt to maintain the discharge pressure at precisely the same pressure required by the operator, i.e., 43 PSIG, regardless of the flow rate demand by the steam power plant. On the left side of Fig. 55, the Worthington pump's discharge pressure did not remain constant, but keep fluctuating, depending upon the new operating points created by the control valve's changeable opening. The power consumption of the Worthington pump changed slightly (see Fig. 53). That was because the operating point keep moving along the pump's curve due to the control valve's varying opening position changing the system's resistance curve's slope. This is the job of the control valve. The control valve changes the system's resistance curve by increasing its frictional losses "minor losses" (see Eq. 15).

This operating principle is different in the case of the variable speed pumps operating at a constant discharge pressure of 43 PSIG. Whenever the operating point changed because of the control valve changing to keep the water level in the DA tank constant, the variable speed pumps sped up to keep the set point pressure constant. The increase in speed caused the variable speed pumps to burn more energy according to the Affinity Laws (see Eq. (19) [68]). From Eq. (19), the power consumption increased when the pumps' speed increased. That explains the peaks in the power consumption of the Grundfos pumps in Fig. 53.

$$\frac{BHP_1}{BHP_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{19}$$

For Test #2 in Case 1, using the average values of the Worthington pump's flow rate and discharge pressure (175.39 GPM and 41 PSIG) to read the power consumption from its performance curve (Appendix A1) gave approximately 6.6 HP (4.9216 kW). That is approximately 3.89% less than the average recorded power consumption (5.121 kW). Using the Grundfos pumps' performance curve (Appendix A2) by applying the recorded average flow rate and discharge pressure of 175.87 GPM and 42.93 PSIG, the power consumption was approximately 5.45kW (7.308 HP). That is lower than the average recorded value (5.796 kW) by approximately 5.97%. These comparisons showed great agreement between the recorded values and the theoretical values read from their corresponding pump performance curves. From the Worthington pump curves (Appendix A1), the pump efficiency at a flow rate of 175.39 is roughly 67%. The system efficiency is 58.6%. Using the average flow rate (175.39 GPM), discharge pressure (41.09 PSIG) and the pump efficiency information in Eq. (18), with the average inlet pressure of 1.8 PSIG subtracted from the discharge pressure, the power consumption obtained from Eq. (18) was 6.858 HP (5.118 kW). This calculated power consumption was nearly the same (0.07% different) as the recorded power consumption of 5.1216 kW. Again, this shows the great agreement between the calculated power consumption and the recorded power consumption for this situation.

Following the same procedure in order to calculate the Grundfos pumps' power consumption using Eq. (18), the system efficiency from the performance curve (Appendix A2) was 58.1% when applying the average recorded values (175.87 GPM and 42.93 PSIG). The system power consumption using Eq. (18) was 7.2 HP (5.369 kW). This calculated power consumption was lower than the recorded value (5.796 kW) by roughly 7.36%. Overall, the calculated power consumption using pump performance curves and Eq. (18) in Case 1, Test #2 was still within an acceptable tolerance range of ± 10%.

It is true that the efficiency of each pump was taken from the pump curves and was subject to error when reading the respective pump curves' efficiency. However, one could conclude that the Worthington pump was running slightly more efficiently than the Grundfos pumps. It might be that this comparison does not put the pumps on the same basis, but still such a comparison can be discussed. The comparison can be justified by examining the Worthington pump's power consumption, which was 13.16% lower than the Grundfos pumps' consumption in Case 1, Test #2. These results give a comparison for both pumps when they were producing almost the same flow rate, but different discharge pressures. The discharge pressure of the Grundfos pumps was higher by 1.84 PSIG.

Similar to Section 3.1.1, the Worthington pump was running on the left side of the BEP [the BEP of the Worthington is 77.5% (see Appendix A1)]. The pump was not working in the rated operating region (80%-110%) of the BEP [4], nor was it working in the preferred operating region of 70%-120% of the BEP [5] (see pump performance curves in Appendix A1 for more information). Therefore, the pump was not sized appropriately to operate at such conditions. In order to re-size the Worthington pump properly, it is recommended to lower the discharge head.

3.2 Case 2

3.2.1 Test #1, March 11, 2015

In the following set of results, the Grundfos pumps were running in the level control mode, based on input from the level sensor installed in the DA tank. In this operating mode, the Grundfos pumps were unable to deliver the condensate water to the vent condenser. Therefore, in order not to overheat the vent condenser when no water was going through its tube bundle, the valve in the pipeline to the vent condenser was turned off. The same process was followed when running the Worthington pump, even though it was capable of lifting the water to the vent condenser. This was done in order to compare the two types of pumps' performance on an equal basis. Each pump ran for three hours in this test.

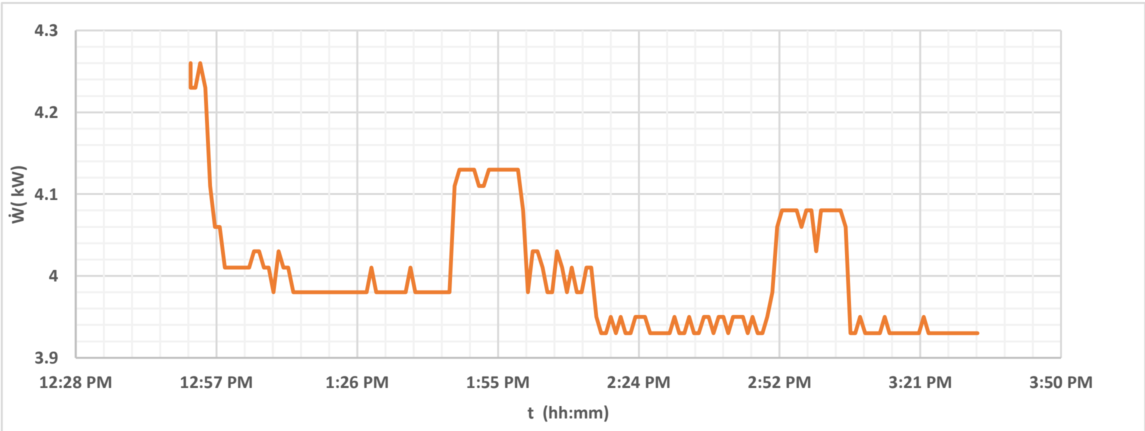


Figure 56: Worthington pump power consumption on March 11, 2015 (Case 2, Test #1)

It can be clearly seen from Fig. 56 how the Worthington pump power consumption was reduced by not providing water to the vent condenser; and the power consumption fluctuated slightly around 4 kW. The average power consumption for the Worthington pump was 3.99 kW over the three hour time span.

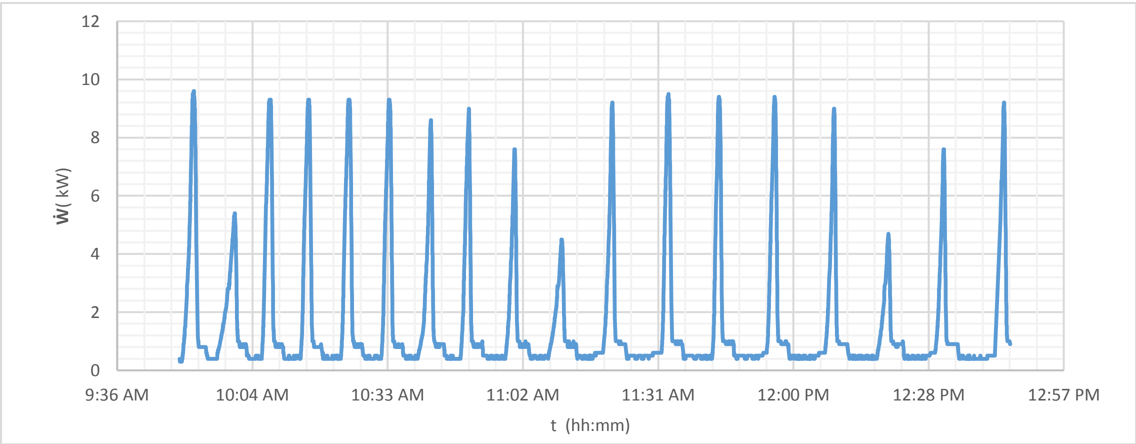


Figure 57: Grundfos pumps' power consumption on March 11, 2015 (Case 2, Test #1)

Figure 57 shows the power consumption of the Grundfos pumps when running in level control mode. The power consumption fluctuated in a periodic manner. The power consumption peaked many time due to the pumps' sensitivity to changes in the water level in the DA tank. Once the water level decreased 1% below the set point of 52%, the pumps reacted as fast as possible to maintain the water level in the DA tank at the precise set point. Such fast reaction caused the power consumption to peak as in Fig. 57. However, these peaks lasted for a short period of time, ranging from 9 to 40 seconds, depending upon the amount of water taken from the DA tank. Then consumption dropped steeply to the lowest level. Even though these peaks existed, the Grundfos pumps consumed much less energy than the Worthington pump. The average power consumption for this operation mode was 1.592 kW. That is 2/3 of the Worthington pump's consumption.

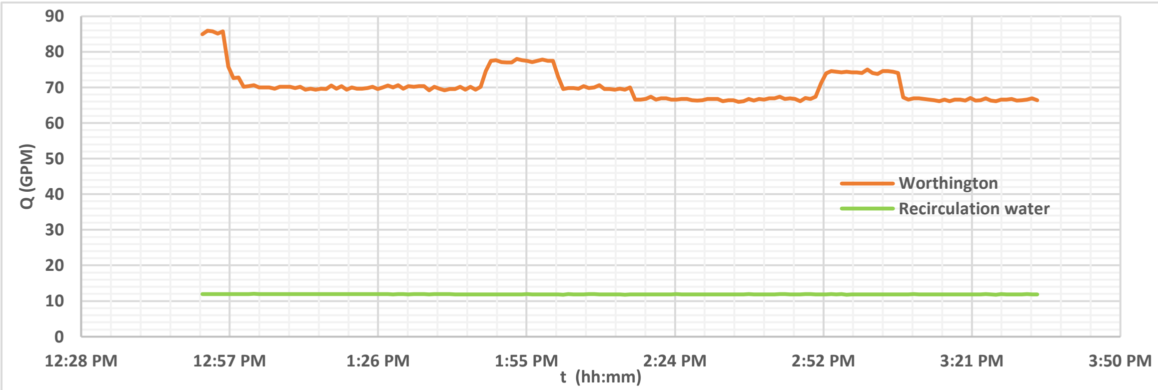


Figure 58: Worthington pump and recirculation flow rates on March 11, 2015 (Case 2, Test #1)

Figure 58 shows the Worthington pump's flow rate when just providing the water to the DA tank. However, because the discharge pressure was high, steam power plant personnel decided to open the recirculation line (Fig. 17) to relieve the pressure from the system, as shown by the 12 GPM recirculation flow in Fig 58. The average flow rate for the Worthington pump was 70.22 GPM, and the average recirculation flow rate was 11.88 GPM.

The Grundfos pumps' flow rate was not recorded for this case because the pressure sensor in the pumps' controller was not selected as the primary sensor. Therefore, the PC-Tools program was not reading the pumps' discharge pressure and flow at that time. The pumps' controller can read the flow rate only when the pressure sensor is in use. The pressure sensor was not selected as a primary sensor by mistake in the first test when changing the pumps' operating mode from constant pressure to level control mode. However, the following sets of tests results had complete information for this kind of operation.

Figure 59 presents the Worthington pump's discharge pressure when the vent condenser was turned off. The Worthington pump's discharge pressure was much higher than that when the vent condenser valve was open. The average discharge pressure for the Worthington pump was 48.65 PSIG.

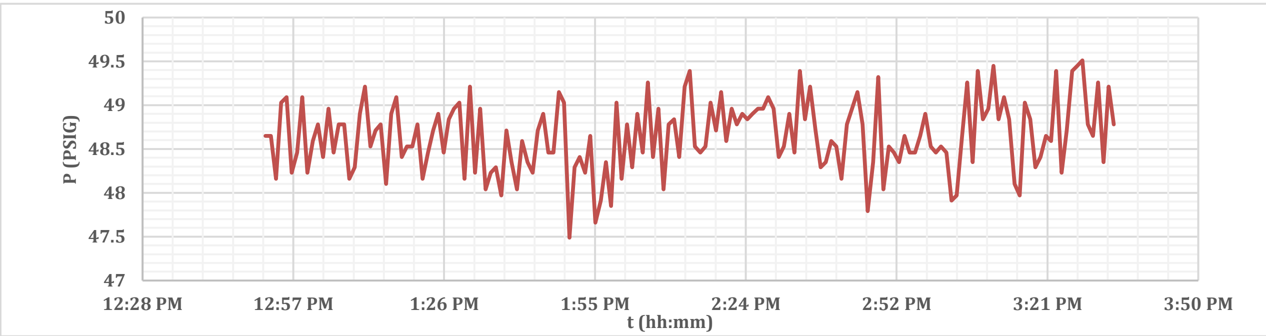


Figure 59: Worthington pump discharge pressure on March 11, 2015 (Case 2, Test #1)

Using the average total flow rate [from adding together the recirculation and the discharge flow rates] and the discharge pressure of the Worthington pump, the power consumption found using the pump performance curve (Appendix A1) was approximately 5.2 HP (3.877 kW). The value obtained from the curve was about 2.8% lower than the recorded real value of 3.99 kW. This result shows great agreement between the pump performance curve information and the data gathered from pump instrumentation.

The Worthington pump's efficiency from the performance curve was approximately 41% due to the low flow rate, and the combined efficiency for the system was 35.87%, which was poor due to operating in the low flow rate region. The calculated power consumption from Eq. (18) was 6.25 HP (4.665 kW), which was 16.9% more than the recorded power consumption. If this value were accurate, the Worthington pump would have been exceeding its design power. This is not reasonable because the pump would be performing better in its "old" condition than when it was new. A close examination of the system's operation was needed. The 1" recirculation line was connected directly to the pump's discharge line and showed a pressure of 48.65 PSIG; and it recirculated the water to the pump's suction line.

It was assumed that the recirculation line increased the inlet pressure of the pump, making it work even less to increase the discharge head. According to the calculations, the inlet pressure was assumed to have increased to 7 PSIG [from 1.8 PSIG] due to the recirculation line high pressure. Therefore, after changing the inlet pressure to 7 PSIG [from 1.8 PSIG], the theoretical power consumption obtained from Eq. (18) for the Worthington pump was found to be 5.56 HP (4.147 kW). This power consumption is 3.9% more than the recorded value (3.99 kW), which is within an acceptable range of ± 10%.

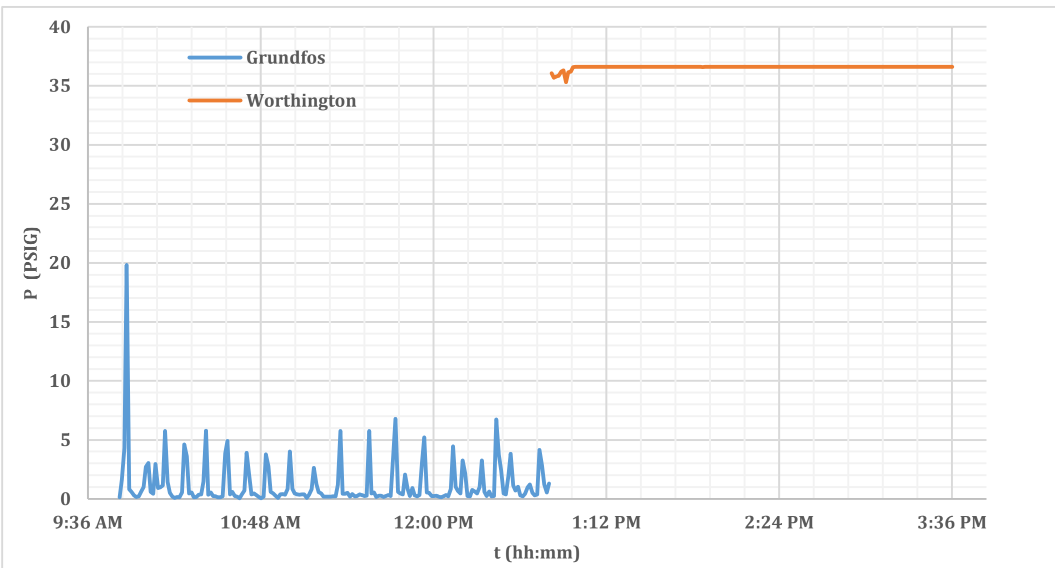


Figure 60: Grundfos pumps and Worthington pump pressure drop across the control valve on March 11, 2015 (Case 2, Test #1)

Figure 60 shows the pressure drop across the control valve for both types of pumps. It can be seen that the pressure dropped almost to zero for the Grundfos pumps when the control valve was fully open, and the average pressure drop was 1.2 PSIG for the Grundfos pumps. 1.2 PSIG was because of the peak flow rates [seen in Test #2, Fig. 64] that caused the pressure drop across the control valve to be as shown in Fig. 60. The first peak on the left side in Fig. 60 occurred because the control valve was not yet fully open, and the Grundfos pumps were running in level control mode. They were speeding up so that the water could reach the DA tank. Thus, this peak was the largest. After the control valve fully opened, no high pressure peaks occurred. On the right side of Fig. 60 is the pressure drop for the Worthington pump when the control valve was in use. One can see the difference in the pressure drop created by the control valve when the vent condenser was shut off from the system. However, the pressure drop shown in Fig. 60 was not the actual value, but it should have been greater. The differential pressure sensor was only able to measure a pressure drop up to 2.5 bar (36.259 PSIG). This is the reason that the recorded pressure curve is almost perfectly flat. The average recorded pressure drop in this case was running was 36.577 PSIG, but should have been higher.

3.2.2 Test #2, April 1, 2015

For this test, the Grundfos pumps ran for one hour, and the Worthington pump ran for just half of an hour because of the high pressure associated with running this pump without using the vent condenser. This high pressure caused pipe leakage in some places. Even though maintenance on the pipes was needed, because the steam power plant has to be fully operational for 24 hours daily, no maintenance could be performed until May 19 of each year [for one week]. The plant is shut down for that week, so that the steam power plant's staff can perform necessary maintenance.

Figure 61 gives the power consumption of the Worthington pump during the afore-mentioned half hour. The average power consumption was 3.99 kW. In contrast, Fig. 62 shows that the Grundfos pumps consumed a great amount of power at times when running in level control mode with the vent condenser turned off.

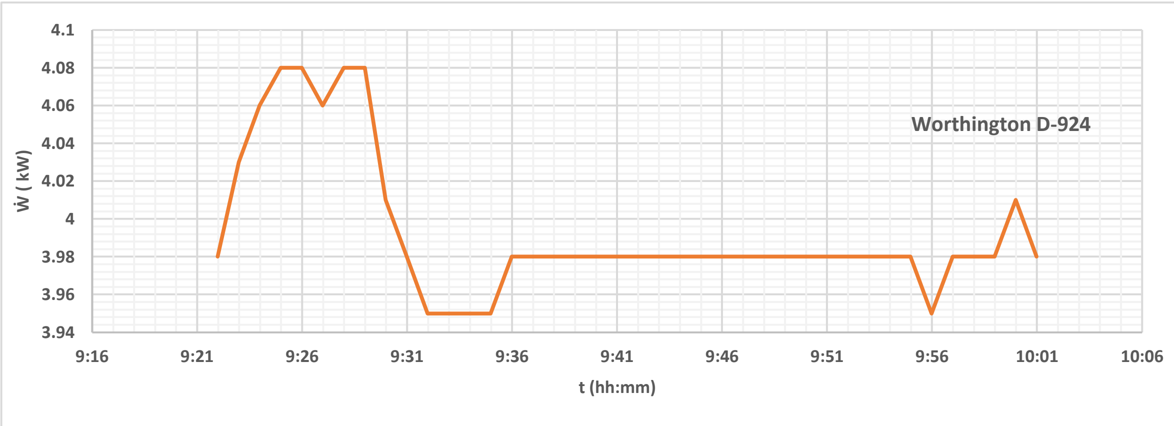


Figure 61: Worthington pump power consumption on April 1, 2015 (Case 2, Test #2)

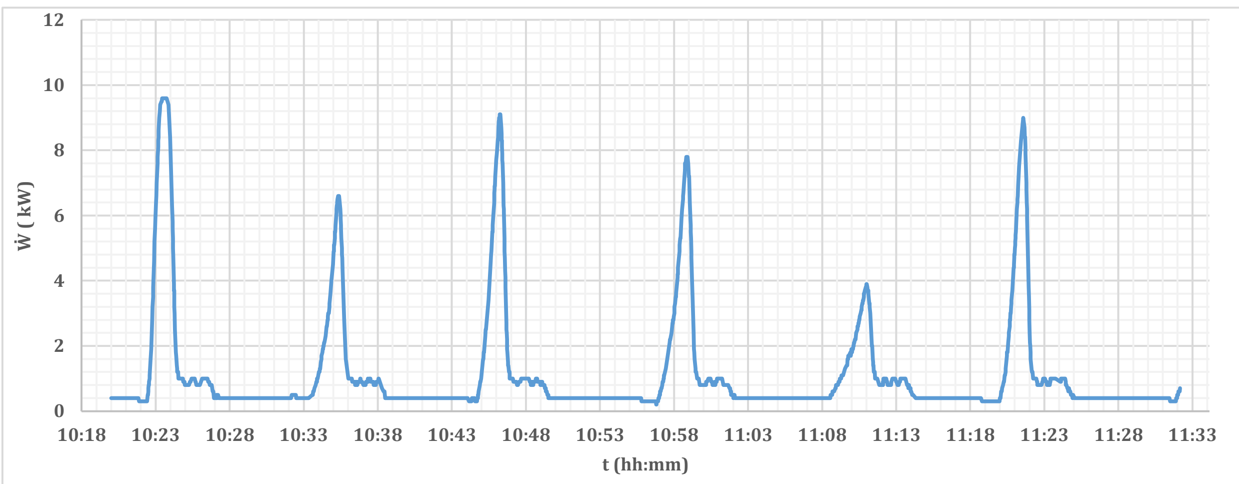


Figure 62: Grundfos power consumption on April 1, 2015 (Case 2, Test #2)

Figure 62 shows fluctuations that are similar to those of Fig. 57. Again, this was because of the pumps' sensitivity to the water level dropping below the set point of the DA tank. In the "troughs" between peaks in Fig. 62, there was fairly constant power consumption which lasted for approximately 6-7 minutes. That was due to the fact that the pumps shifted from normal operation to minimum work performance in order to save energy. The average power consumption for the Grundfos pumps was 1.16 kW. That means the Grundfos pumps were able to run using 29% of the Worthington pump's power consumption.

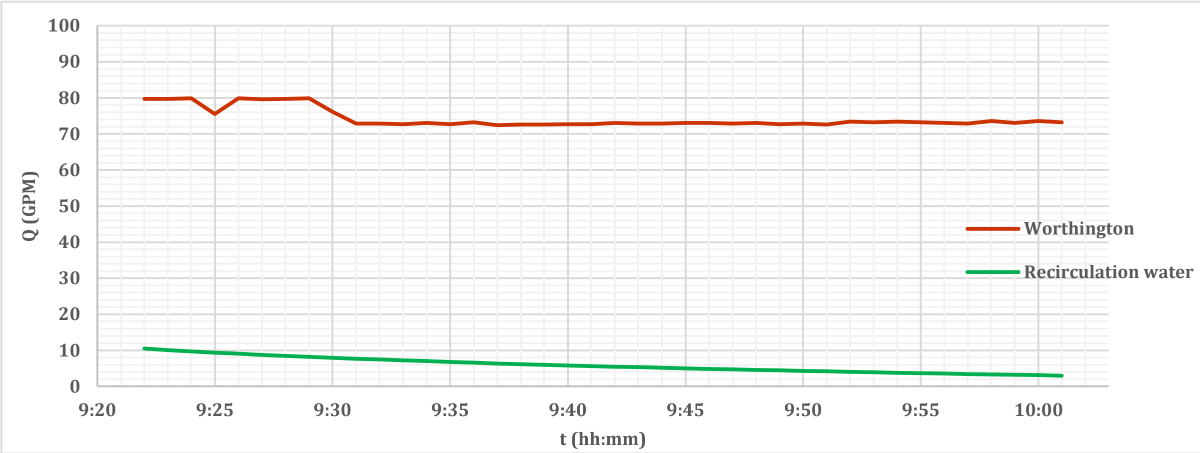


Figure 63: Worthington pump and recirculation flow rates on April 1, 2015 (Case 2, Test #2)

Figure 63 presents the condensate water flow rate required to feed the DA tank while the recirculation line was open. In Fig. 63, the recirculation flow rate started at 10.5 GPM, then uniformly decreased to 2.99 GPM. This was the case every time that the data was gathered. Even though the magnetic flow meter installed in the recirculation line (see Fig. 17) displayed a fairly constant flow rate during the time of the experiment, the data acquisition system recorded lower values than those given by the magnetic flow meter after a short time from the beginning of each test. The data acquisition system's fourth channel, into which the recirculation line's data cable connector was plugged, was reset and recalibrated, i.e., the maximum and minimum reading range were re-input. However, the data acquisition system still followed the same scenario every time the experiment was carried out. Therefore, in order to have a roughly accurate calculation, the recirculation line's average flow rate was taken only during the first few minutes of testing time, because the reading at the beginning of the test was similar to that recorded by the data acquisition system. The average flow rate of the Worthington pump for this time period was 74.32 GPM, and the average flow rate in the recirculation line was 5.934 GPM. However, when taking the recirculation line flow rate average only for a time period of four minutes from the start of the test, the average flow rate was 9.7 GPM. Therefore, the total average flow rate through both the Worthington pump and the recirculation line was 84 GPM, which will be used in the calculations.

In Case 2, Test #2, the PC-Tools software was able to record the Grundfos pumps' flow rate because the discharge pressure sensor was selected to be one of the sensors that fed the Grundfos pumps' controller with discharge pressure information. As discussed previously, the Grundfos pumps' controller can read the flow rate only when the pressure sensor is in use. Therefore, the PC-Tools software was able to record the flow rate from the Grundfos pumps' controller as shown in Fig. 64. The recirculation line was completely closed when the Grundfos pumps were running in level control mode.

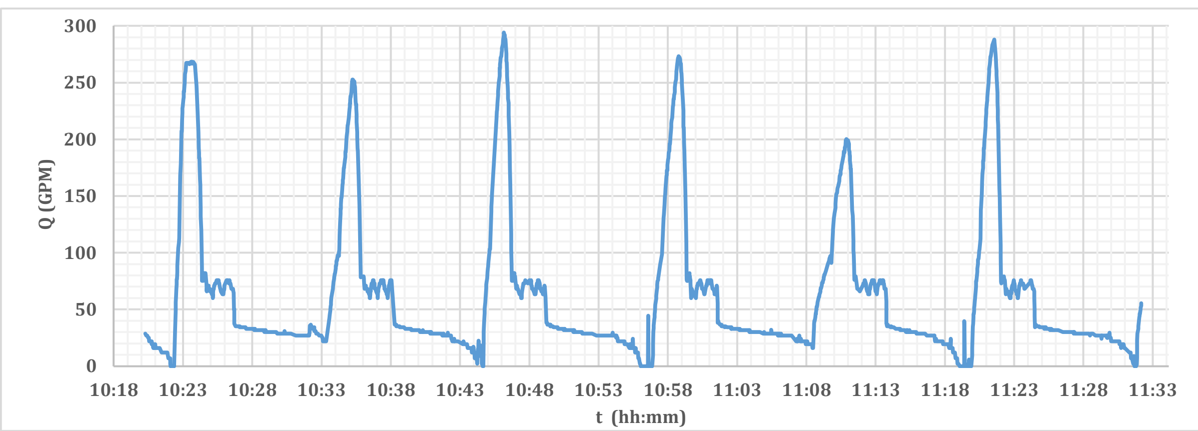


Figure 64: Grundfos pumps flow rate on April 1, 2015 (Case 2, Test #2)

As is evident from Fig. 64, the flow rate also peaked after the times when the water level in the DA tank decreased below the 52% set point. The time span for the first peak when the flow rate reached 247 GPM to 268 GPM was 56.76 second. This means that a significantly high flow rate reached the DA tank during that 56.76 second period. Recalling the DA tank's operating principle, there should be enough steam to heat up that large quantity of water reaching the DA for only approximately one minute. Otherwise, the temperature of the water stored in the DA tank decreased; and, as a result, the boiler efficiency decreased correspondingly. However, when examining the last peak, it was found the average flow rate was 277.442 GPM, but over a time period of 38.87 seconds. This means that 179.78 gallons reached the DA tank during this relatively small time period. This was one of the disadvantages of these peaks. The other disadvantage was the high power consumption which caused the pumps to work hard to return the water level in the DA tank to the 52% set point. The average flow rate over 1 hour for the Grundfos pumps was 61.57 GPM.

Figure 65 shows the discharge pressure for both types of pumps on April 1, 2015. It is obvious that the Worthington pump provided a high pressure to overcome the system frictional head, especially the local frictional losses produced by to the control valve. This fact can explain why the Worthington pump was able to provide enough pressure head to lift the water to the vent condenser, because of the control valve flow restriction. This restriction caused the Worthington pump to run in a low flow rate region. Therefore, from the pump curve (see Appendix A1), the developed pressure head was high for low flow rate operation.

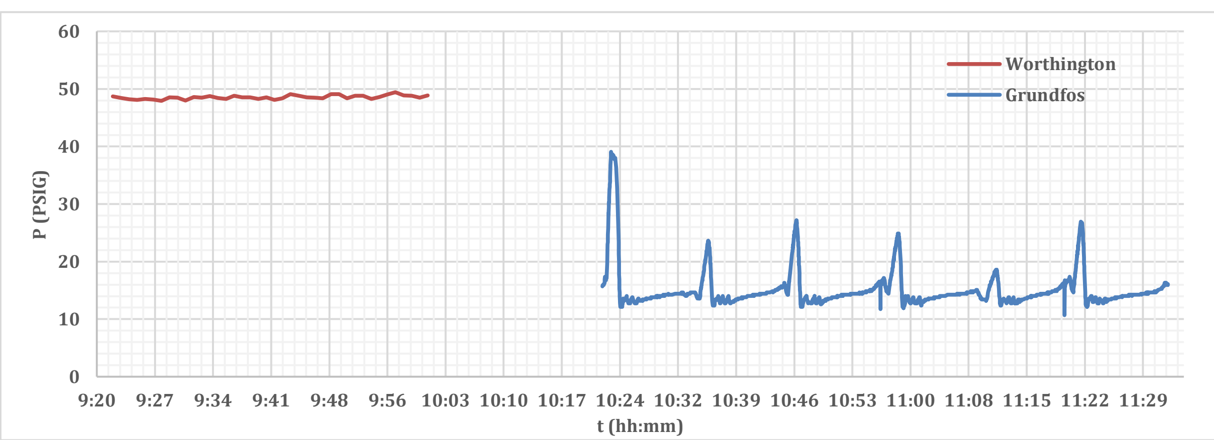


Figure 65: Worthington and Grundfos pumps discharge pressure on April 1, 2015 (Case 2, Test #2)

Again, the initial peak in the Grundfos pumps' pressure shown in Fig. 65 was due to the control valve not being fully open. That forced the pumps to produce a high pressure in order to overcome the control valve's frictional head losses so that condensate water could reach the DA tank. The average discharge pressure was 16.13 and 48.56 PSIG for the Grundfos and Worthington pumps, respectively.

Using the recorded average flow rate and discharge pressure to obtain the power consumption for Worthington pump from its performance curve (Appendix A1) gave approximately 5.2 HP (3.877 kW), which is similar to that (3.877 kW) from Test #1 because the flow rate did not change much. The Worthington pump's power consumption obtained from the curve was about 2.83% lower than the recorded power consumption (3.99 kW). However, when using Eq. (18) to find the theoretical power for the Worthington pump. The Worthington pump and motor efficiency was 35.87%. The power consumption was 5.456 HP (4 kW), which is about 0.25% higher than the recorded value. This gives an indication that the assumption made (the inlet pressure was 7 PSIG instead of 1.8 PSIG due to the recirculation line pressure effect on the pump inlet pressure) was reasonable and acceptable.

As for the Grundfos pumps, it was not possible to have recorded discharge pressure and flow rate at the same time in this Test. The pumps' data changed significantly every second. Therefore, in order to have an estimation of the pump efficiency, curve fitting was performed on the increasing part of the first pulse assuming that every other peak behaved in similarly. A third order polynomial equation was found to fit all of the data gathered. See Appendix G for more information. The third order polynomial equation gave reasonable percentage errors as compared to the data recorded for each parameter (flow rate, power and discharge pressure) (see Appendix G). The flow rate and discharge pressure values were used to obtain the power consumption from the pumps' performance curves, and Eq. (18) was used to calculate the theoretical power consumption. Then the recorded power consumption was compared with the two calculated pump power consumption values obtained from the pumps' performance curves and from Eq. (18)). The discharge pressure used in Eq. (18) was the differential discharge pressure and not the total head developed by the pump. The inlet pressure was assumed to be 1.8 PSIG.

All results are shown in Table 6. A negative sign in the error column means that the calculated value was less than the recorded value.

Table 6: Power consumption comparison for the Grundfos pumps (Case 2, Test #2)

#	Time (sec)	Pump &motor efficiency from the pump curve (Appendix A2)	Recorded power consumption (kW)* (Appendix G) (X1)	Power consumption from the pump curve (kW) (Appendix A2) (X2)	Error **	Power consumption from Eq. (18) (kW)** (X3)	Error***
1	7.34	51%	1.63	1.61	-1.22%	1.575	-3.37%
2	17.1	52%	3.55	3.56	0.28%	3.438	-3.15%
3	35.16	50.81%	7.699	7.6 kW	-1.28%	7.336	-4.71%

* See Appendix G for the corresponding values of flow rate and discharge pressure obtained from curve fitting.

** The errors obtain from $Error\ (\%) = \frac{X2-X1}{X1} 100$

*** The errors obtain from $Error\ (\%) = \frac{X3-X1}{X1} 100$

3.2.3 Test #3, April 2, 2015

In this test, the Worthington pump ran for approximately half an hour, and the Grundfos pump ran for one hour**.**

Figure 66 shows the power consumption of the Worthington pump when the vent condenser was off. The average power consumption for the Worthington pump was 3.934 kW for the specified time. Even though there was a savings in the power consumption in this case when running the Worthington pump without having the vent condenser, this saving was not comparable to that of the Grundfos pumps when running in level control mode, shown in Fig. 67.

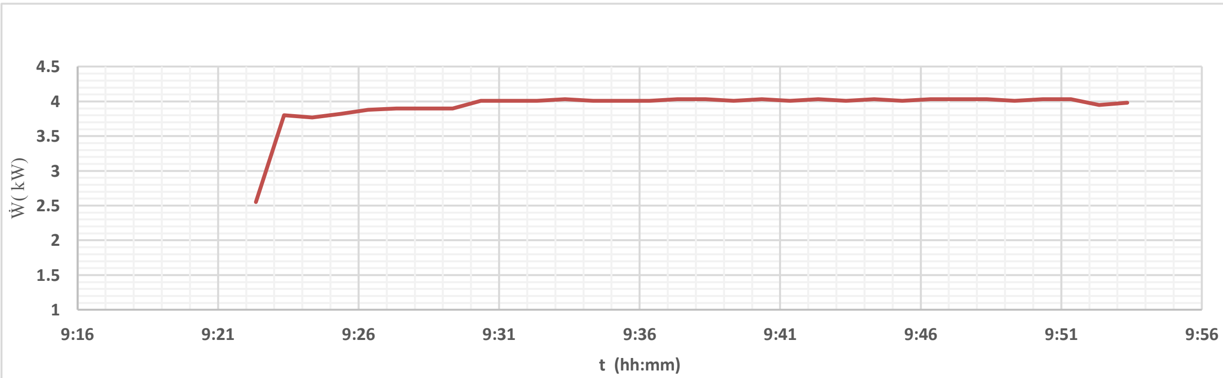


Figure 66: Worthington pump power consumption on April 2, 2015 (Case 2, Test #3)

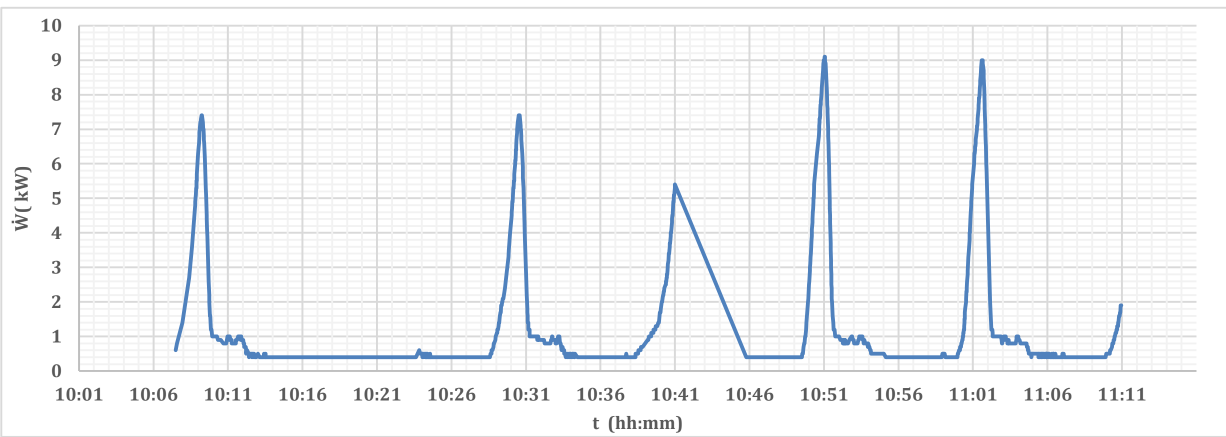


Figure 67: Grundfos pumps' power consumption on April 2, 2015 (Case 2, Test #3)

Figure 67 shows a similar scenario to that previously discussed. Power consumption peaked when the Grundfos pumps tried to return the water level in the DA tank to the set point of 52% (see the discussion of Fig. 41 for more information). However, the average power consumption was significantly lower than that of the Worthington pump. The average power consumption of the Grundfos pumps was 1.619 kW during approximately one hour on April 2, 2015.

When looking at Fig. 68, it can be seen that the Worthington pump's flow rate was similar to that of the test made on April 1. Therefore, the power consumption was also almost the same. Again, the recirculation line flow rate experienced the same problem that was explained with regard to Fig. 63.

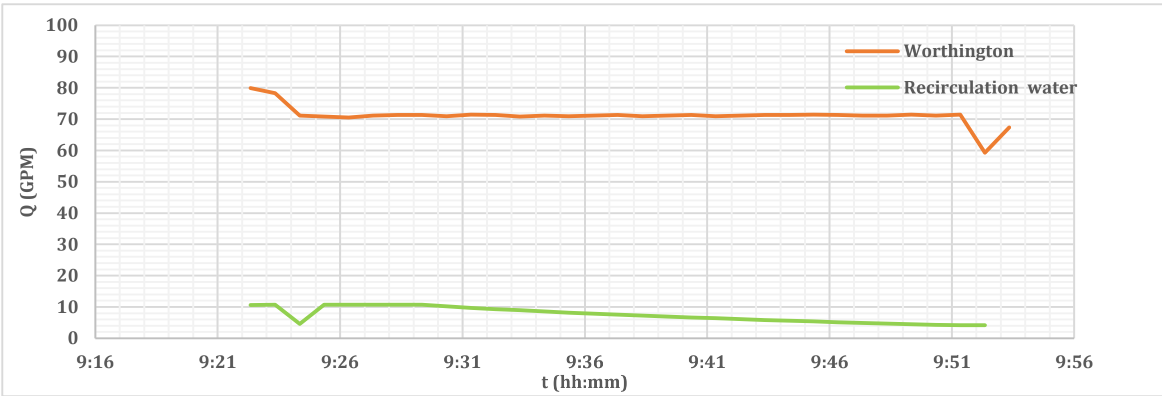


Figure 68: Worthington pump and recirculation flow rates on April 2, 2015 (Case 2, Test #3)

The average flow rate of the Worthington pump was 71.15 GPM. Following the same procedure as discussed for the Test made on April 1 in calculating the average flow rate in the recirculation line, that average was found to be 9.88 GPM for the first ten minutes as compared to the overall average flow rate of 7.282 GPM for thirty minutes. Therefore, the total average flow rate was computed as 71.15+9.88 = 81.03 GPM.

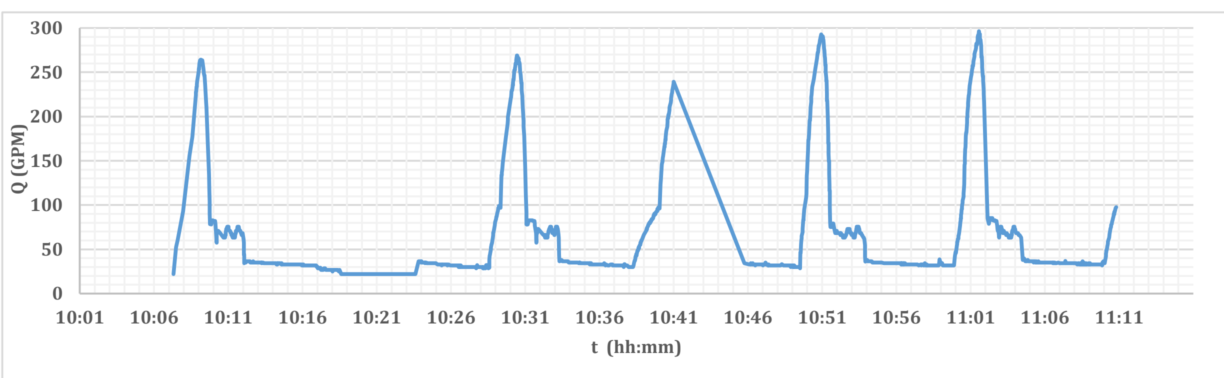


Figure 69: Grundfos pumps' flow rate on April 1, 2015 (Case 2, Test # 3)

The average flow rate of the Grundfos pumps was 77.94 GPM over one hour of operation. In Fig. 70, the discharge pressure for both pumps shows how the system pressure was relatively low when the Grundfos pumps were running on level mode.

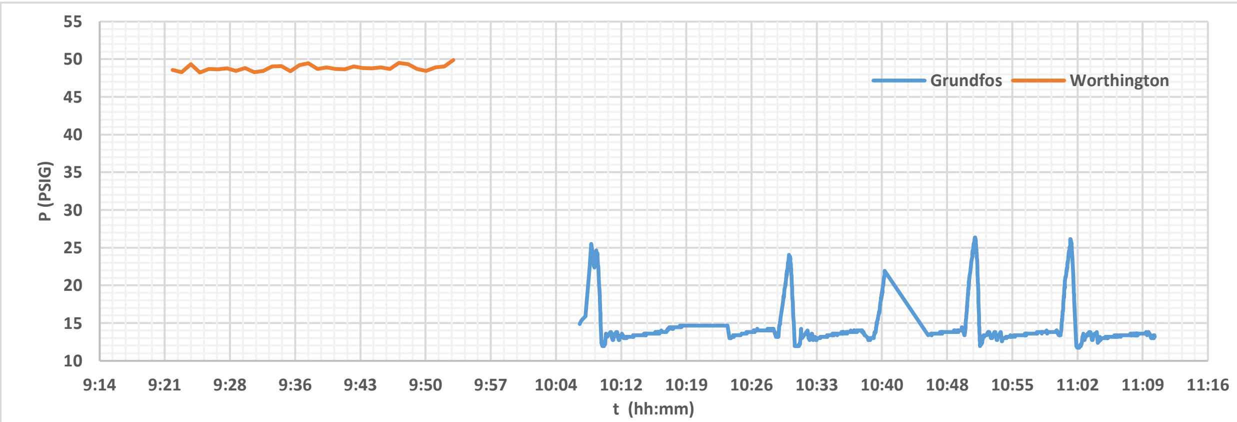


Figure 70: Worthington pump and Grundfos pumps' discharge pressure on April 2, 2015 (Case 2, Test # 3)

The average discharge pressure of the Worthington pump and Grundfos pumps was 48.8 PSIG and 14.98 PSIG, respectively.

When using the average flow rate and discharge pressure of the Worthington pump for its pump performance curve, it was found that the Worthington pump was consuming 5.2 HP (3.877 kW) at that operation point. That means the power consumption from the curve was 1.44% lower than the recorded value (3.934 kW). That might be because the Worthington pump was "old". Using the same performance curve to obtain the pump efficiency, the pump part of the system efficiency was found to be 40%. Therefore, the overall efficiency can be found by considering the pump's motor efficiency of 87.5%. Therefore, the overall pump and motor efficiency was 35%. The recorded data gave a power consumption, when using Eq. (18), equal to 5.6 HP (4.175 kW). This theoretical power consumption was 6.12% greater than the recorded value (3.934 kW). Again, that might be justified by assuming that the inlet pressure was more than the assumed value of 7 PSIG.

The Grundfos pumps' efficiency in this test was much easier to determine since the PC-Tools software recorded the data so as to find a flow rate and discharge pressure at exactly the same time. Therefore, in order to determine the Grundfos pumps' efficiency in these tests, three operating points were selected at different times on the increasing curve of a pulse [of Figs. 67, 69, and 70] and analyzed in order to calculate the Grundfos pumps efficiency as shown in Table 7. A negative sign in the error column means that the calculated value is less than the recorded value.

Table 7: Power consumption comparison for the Grundfos pumps (Case 2, Test #3)

Time	**Flow Rate (GPM)**	**Discharge Pressure (PSIG)***	**Recorded power consumption (kW) (X1)**	**Pump & Motor Efficiency from Pump Curve (X2)**	**Power Consumption from Pump Curve (kW) (X3)**	**Error****	**Power Consumption Obtained from Eq.(18)**	**Error*****
10:08:04	92.90	15.896	1.4	49.8%	1.32	5.71%	1.14	18%
10:09:00	236.43	22.785	5.5	41.5%	5.77	-4.9%	5.2	-5.4%
10:09:15	263.73	24.453	7.1	39.4%	7.27	-2.39%	6.597	7%

* The discharge pressure used in Eq. (18) was the differential discharge pressure and not the total head developed by the pump. The inlet pressure was assumed to be 1.8 PSIG.

** See Table 6.

*** See Table 6.

3.2.4 Test #4, April 9, 2015

In this test, the Worthington pump ran first for one hour, then the Grundfos pumps took over the work for another one and a half hours.

As discussed previously regarding Fig 41, the Time Integral (T_i) of the Grundfos pumps had been selected as 2 seconds. However, in this test, T_i was changed to 1.5 seconds. That made the Grundfos pumps even more sensitive to any change in the water level in the DA tank. In other words, if the water level decreased below the set point of 52%, the Grundfos pumps sped up faster than for the previous tests. This change was made in order to keep the water level in the DA tank nearly the same as the set point. However, this goal was not met. The result from this test was that there were fewer peaks. Also, the water level did not fall below the 50%, as compared to the other tests (see Fig. 41). This is because the pumps were more sensitive and did not let the water level drop below 50% as shown in Fig. 71.

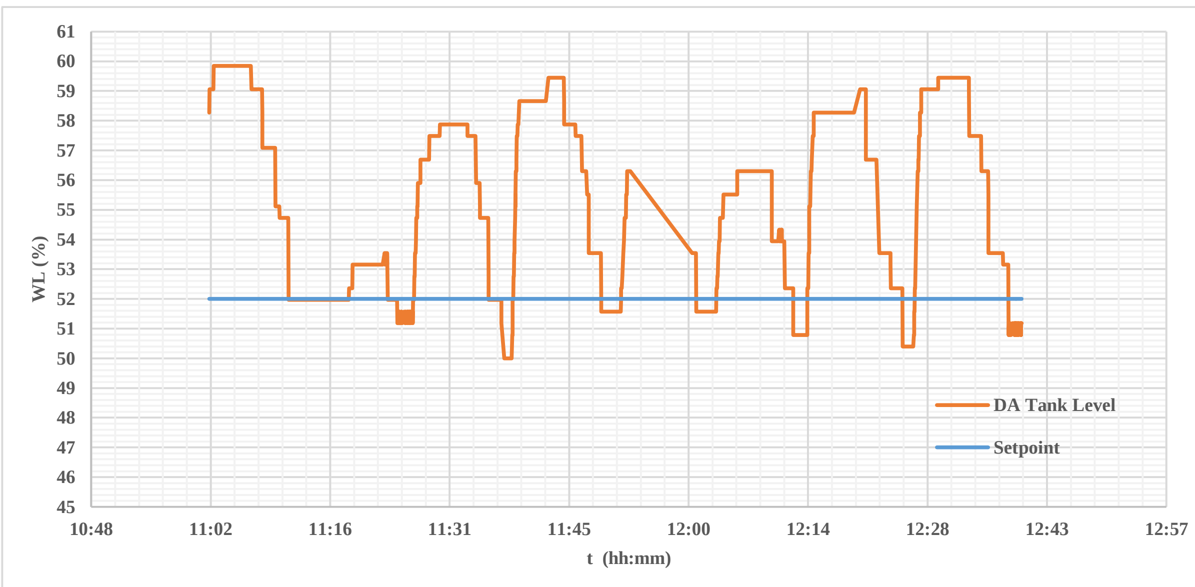


Figure 71: Water level in the DA tank vs. the set point while running Grundfos pumps (Case 2, Test # 4)

The water level was high when the Grundfos pumps started because the Worthington pump ran before the Grundfos pumps. Therefore, the Grundfos pumps were attempting to drop the water level back to the set point as shown at 11:02 in Fig. 71.

Based on the data shown in Fig. 72, the average power consumption of the Worthington pump was 3.99 kW for one hour of operation, which is same power consumption as for the first three tests.

The average power consumption of the Grundfos pumps was 1 kW (Fig. 71) for the same operation time as the Worthington pump. The Grundfos pumps' power consumption was lower because the required flow rate was lower in this test.

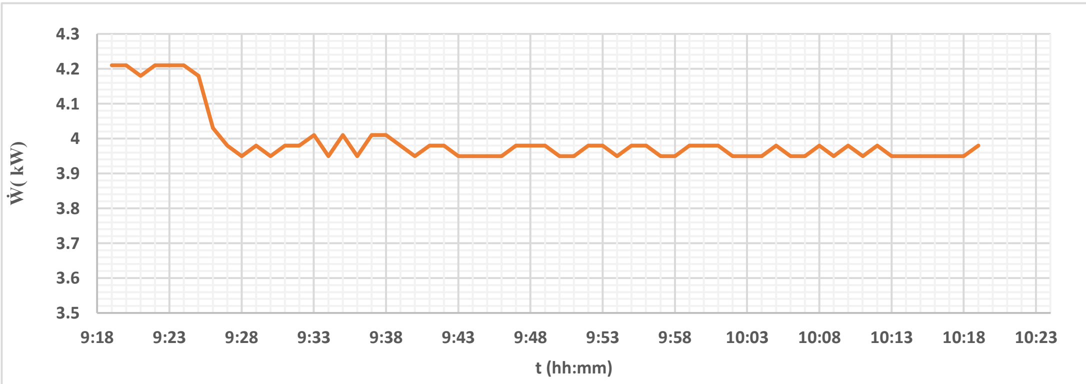


Figure 72: Worthington pump power consumption on April 2, 2015 (Case 2, Test # 4)

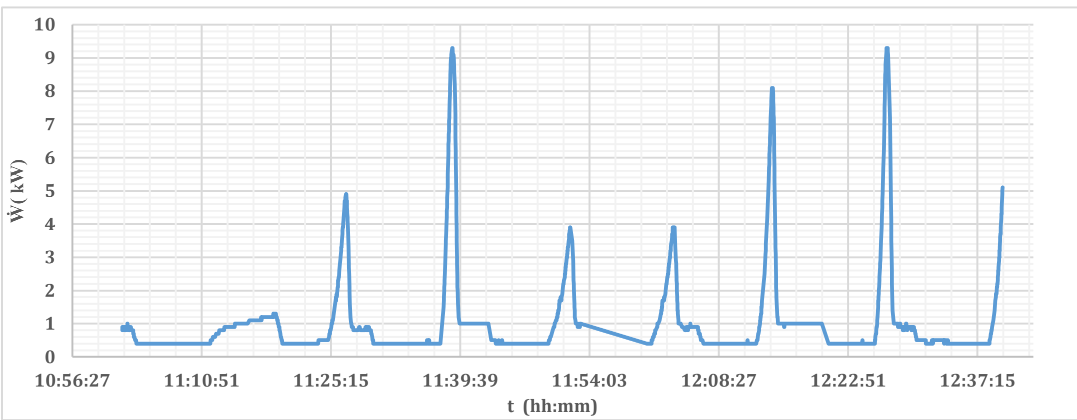


Figure 73: Grundfos pumps' power consumption on April 9, 2015 (Case 2, Test # 4)

The Worthington pump flow rate was the same as for the flow rate tests shown in Figs. 63 and 68. That is because the weather was not cold and did not change during the days when the data was gathered. The average flow rate of the Worthington pump was 70.42 GPM (Fig. 74); and the recirculation line flow rate was fairly constant at 10 GPM displayed on the Siemens flow meter. However, because of the data acquisition problem which showed that the recorded recirculation flow rate drop with time, the flow in the recirculation line dropped from 10 GPM to almost zero exactly at 10:13 in Fig. 74. However, by taking only the first five minutes of recirculation flow rate data in the Fig. 74, the average recirculation line flow rate was 9.21 GPM. Therefore, the total average flow rate was 79.63 GPM, including only the first five minutes of recorded recirculation flow rate.

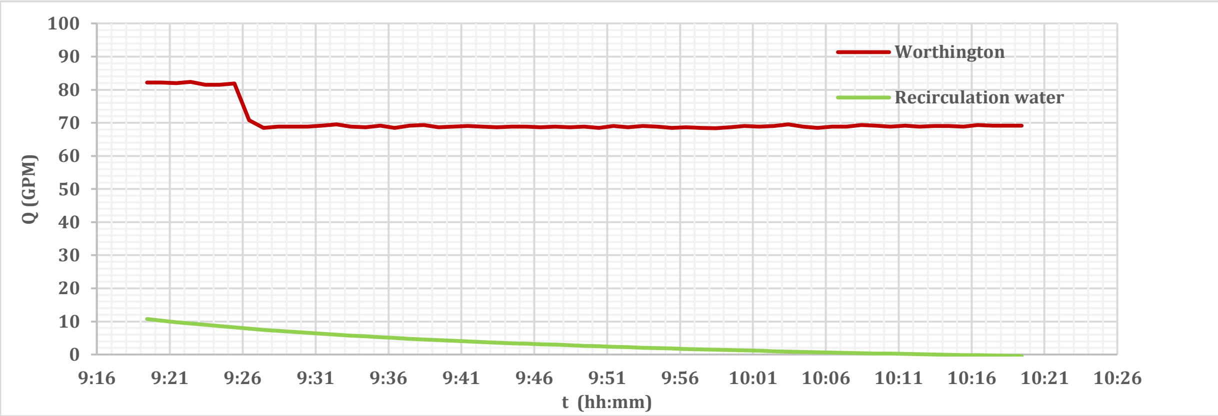


Figure 74: Worthington pump and recirculation water flow rates on April 2, 2015 (Case 2, Test # 4)

The Grundfos pumps' flow rate followed the same fluctuating behavior as for the previously discussed tests even though the response time (T_i) had been changed. Most noticeable in Fig. 75 was that the number of the relatively small peaks increased, and the number of large flow rate peaks decreased.

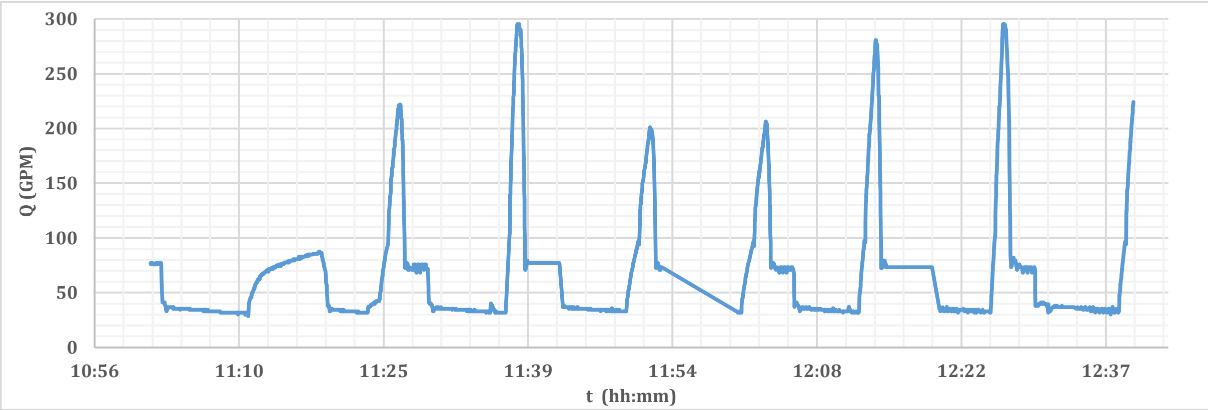


Figure 75: Grundfos pumps' flow rate on April 9, 2015 (Case 2, Test #4)

Again, the recirculation line was closed when the Grundfos pumps ran, so that all of the flow moved into the DA tank. The average flow rate of the Grundfos pumps was 65.2 GPM for the specified time in Fig. 75.

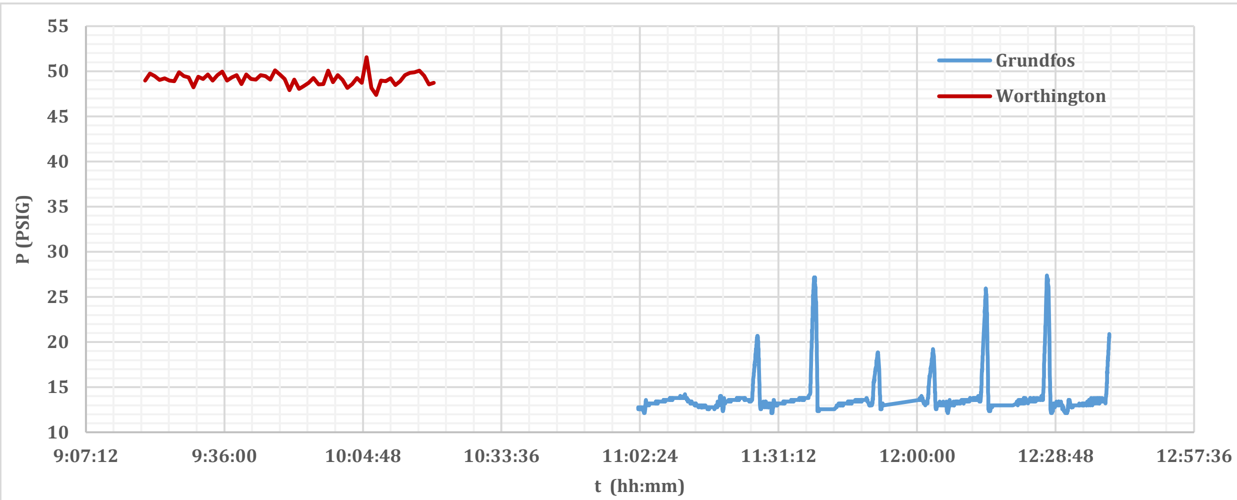


Figure 76: Worthington pump and Grundfos pumps' discharge pressure on April 9, 2015 (Case 2, Test #4)

In Fig. 76, the average discharge pressure of the Worthington pump was 49.13 PSIG for the specified time, while the average discharge pressure of the Grundfos pumps was 13.83 PSIG.

Lastly, when comparing the recorded power consumption of the Worthington pump with that given in its performance curve (operating point plotted by using the recorded average flow rate and discharge pressure) the pump curve showed that the power consumption was 5.12 HP (3.817 kW). That is approximately 4.3% lower than the recorded value of 3.99 kW. Again, one must consider the pump's age that may have caused the pump to consume a little more energy. The pump efficiency was exactly 40%, obtained from the Worthington pump performance curve (Appendix A1). Therefore, the overall system efficiency was 35%. Using Eq. (18), the computed Worthington pump power consumption was 5.59 HP (4.168 kW), about 4.46% higher than the recorded value. Even though the power consumption obtained from Eq. (18) (4.168 kW) was higher than the recorded power consumption (3.99 kW), the calculated error (4.46%) is in the acceptable range ±10%.

In order to compare the Grundfos pumps' recorded power consumption with that calculated theoretically, same process was followed as was done in Test # 3. Again, PC-Tools provided the ability to find more than one operating point exactly at the same time as shown in Table 8. Where the negative sign in the error column means that the calculated value is less than the recorded value. The operating points were taken at different times in order to examine the pumps' performance over a wide range of time.

Table 8: Power consumption comparison for the Grundfos pumps (Case 2, Test #4)

Time	Flow Rate (GPM)	Discharge Pressure (PSIG)*	Recorded Power Consumption (kW)	Pump& Motor Efficiency from Pump Curve	Power Consumption from Pump Curve (kW)	Error**	Power Consumption Obtained from Eq. (18) (kW)	Error***
11:28:24	75.73	12.763	0.9	46%	0.935	-3.88%	0.785	-12.7%
12:12:59	77.49	13.604	0.9	46.8%	1	-11.11%	0.85	5.5%
11:51:01	143.53	15.693	2.1	47.5%	2.08	- 0.95%	1.82	- 13.3%

* The discharge pressure used in Eq. (18) was the differential discharge pressure and not the total head developed by the pump. The inlet pressure was assumed to be 1.8 PSIG.

** See Table 6.

*** See Table 6.

In conclusion, the power consumption obtained from the sensors of the Worthington pump and Grundfos pumps showed reasonable comparisons to their equivalent values found from pump performance curves in Appendix A (A1 and A2). Therefore, the recorded power consumption can be used in estimating the annual energy consumption as part of the total costs of the LCCA for each pump, so that the result can help to determine the most cost effective pumping system.

According to Tests 2, 3 and 4, the Grundfos pumps' discharge pressure was 16.13, 14.98, and 13.83 PSIG, respectively. Those pressures were too low to achieve the required discharge pressure for the vent condenser at point A (~28 PSIG from Appendix E3 calculations for Worthington pump). See Fig. 45 and Section 2.3.3 for pressure drop calculations. Therefore, in these tests, the Grundfos pumps were unable to lift the water to the vent condenser.

Chapter 4: Life Cycle Cost Analysis

4.1 Pumps Life Cycle Analysis

The LCCA discussed in Section 2.4 was implemented using the BLCC5 software downloaded from DOE [36]. Two LCCAs were carried out, one for Case 2 (Section 3.2) and the second for the vent condenser (Section 2.3.2.1), in order to determine savings from using the variable speed pumps and the vent condenser. For Case 1, the Grundfos pumps ran at constant discharge pressure and the Worthington pump ran normally; and the flow for both was controlled by the regulator valve (control valve). Therefore, the Grundfos pumps were emulating the Worthington pump's work which gave little opportunity for the Grundfos pumps to save energy. However, Case 1 was presented so that an overview of the two pumps types' performance could be examined and analyzed in order to have a knowledgeable background of the two pump types when running at fairly constant discharge pressure.

In this project, data for average power consumption was not taken for all months in the year. Therefore, an estimate had to be made in order to predict the power consumption in those months for which the data was not available. This estimate was based on the principle that the power consumption is directly related to the amount of condensate water supplied [10]. The condensate water supplied by the pumps is almost the same as the steam generated by the steam power plant's boilers. The boilers' blowdown is not considered in this calculation because no flow meter was installed to measure the boiler blowdown flow rate, and this flow is considered to be small. The boilers' blowdown is important because not all of the condensate water supplied by the pumps is converted to steam; but some small portion of that condensate water becomes blowdown from the boilers. The other factor not considered was varying pump efficiency (see Eq. (13)). The pumps' efficiencies were assumed not to change when supplying condensate water to the DA tank. This assumption was true for the Worthington pump (see Section 3.2) because the Worthington pump's efficiency did not change significantly during the days on which data was gathered. However, the Grundfos pumps' efficiency changed because these pumps were not supplying a constant flow rate. Their flow rate varied corresponding to water level in the DA tank. Moreover, the overall pumps' average efficiency was not easily determined because data was not gathered at equal intervals by PC-Tools.

The arithmetic average of the power consumption data taken for the Worthington pump was used for the average energy consumption of this pump because the data was taken at equal time intervals. On the other hand, a time-weighted calculation was used to find the average power consumption of the Grundfos pumps because the data was taken over varying intervals of time, as discussed in Section 2.3.

In order to calculate the average annual energy consumption using only the data taken for certain days, the power consumption for the month in which the data was gathered was predicted as follows. If the steam generated for the days on which the data was collected within was within 10% of the average steam generated in that month, then the average energy consumption (kWh) of that month was obtained by multiplying that month's total hours by the average power consumption for the days with known energy consumption [10]. Otherwise, daily average energy consumption was estimated by [10].

$$E_p\,(Desired\ Day)(kWh) = E_p(\ Known\ Day)(kWh)\,\frac{S_G\ (Desired\ Day)\ (lb)}{S_G\ (Day\ with\ known\ E_p)\ (lb)} \qquad (20)$$

After finding the estimated month's energy consumption by summing the daily energy consumption for that month, as shown in Eq. (20), the estimated energy consumption for that month was used to determine the energy consumption of the months of the year [10] for which no energy consumption data was available.

The procedure for estimating monthly energy consumption was similar to the daily energy consumption estimation. The monthly average energy consumption can be found from [10]

$$E_p\ (Desired\ Month)(kWh) = E_p\ \ (Known\ Month)\ (kWh)\frac{S_G(Desired\ Month)\ \ (lb)}{S_G\ (Month\ with\ Known\ E_p)\ (lb)} \tag{21}$$

That procedure was followed for both pump types in Case 2 in order to have the monthly energy consumption for the entire year. The estimated annual energy consumption was achieved by simply adding the 12 months' energy consumption of each pump. After calculating the estimated energy consumption for each pump using the approach discussed above, the estimated annual energy consumption was input into the BLCC5 software for the Worthington and Grundfos pumps in order to have the total LCC for each pump separately. See Appendix H for annual energy consumption calculations [10].

The water cost that was used was the BLLC5 software was divided into water consumption in the summer and water consumption in the winter. The power plant has all of the information regarding the amount of water purchased from the City of Lawrence every month. Therefore, the water usage from April through September was assumed to be summer time, and the water usage from October through March was assumed to be winter time [10]. The same water usage was applied for each pump since there was no difference in water usage between the two pumps; and the water usage mainly depended on the demands and the losses in the steam cycle. All of the information required in the LCC analysis was discussed in Section 2.4. However, one must realize that the only differences between the pumps types' costs were the energy expenditures, initial costs and the impeller replacement costs (which was higher for the Grundfos pumps than for the Worthington pump). The other costs, such as water consumption, mechanical seal replacement, and mechanical seal to be purchase price, were considered to be the same for each pump type.

4.1.1 Case 1

In Case 1, the control valve regulated the flow rate to the DA tank. The control valve throttled the flow. This process created fairly high back pressure that was enough to have condensate water reach the vent condenser. Therefore, the steam power plant was able to benefit from recovery of the condensing steam's and non-condensable gases' energy in the vent condenser. The data that was gathered showed that the Worthington pump provided less water to the system at a lower pressure than did the Grundfos pumps because the external temperature was colder during the Grundfos pumps' operation time period than during the Worthington pump's operation time period. That was true for Case 1, Test #1. Then Test #2 showed that, even though the two pump types were being compared for doing approximately the same task, the Grundfos pumps were not able to save energy. They consumed 13.181% more energy (0.675 kW) than the Worthington pump (5.121 kW).

In Case 1, the discharge head provided by the Grundfos pumps was fixed because they were running in pressure control mode. Therefore, the Grundfos pumps provided a varying flow rate; but the discharge head was constant. On the other hand, the Worthington pump provided both varied flow rate and discharge head. Thus, depending on the outside temperature, when the steam demand was high, the Worthington pump provided a high flow rate with the designed discharge head at that flow rate. However, under the same conditions, the Grundfos pumps provided high flow rate and a relatively higher discharge head than the Worthington pump. In conclusion, the comparison should be made on the same day under the same operating conditions so that both types of pumps could be compared on the same basis. Therefore, Case 1 was not suited for performing a comparative LCCA. However, Case 1 is presented in this thesis in order to have an overview as to how the Grundfos pumps functioned when they were running in discharge pressure control mode.

4.1.2 Case 2

In Case 2, both the Grundfos and the Worthington pumps were running to feed the DA tank with the condensate water on the same day during consecutive operating time periods. The Grundfos pumps were running in level control mode. The vent condenser was isolated from the system due to the low discharge head provided by the Grundfos pumps when running in level control mode (see Section 3.2). The power consumption information for Case 2 was taken in March and April of 2015. However, there was no full information of the steam production for the whole month of April, 2015 because this thesis was written in mid-April of 2015. Therefore, the generated steam information was only available for about half of April of 2015. Thus, the total average steam production of April of 2014 was used in order to calculate the annual energy consumption from the months for which no power consumption was recorded (see Eq. (21) and Appendix H).

When comparing the steam production of March 11 with average steam generated in that month, it was found that the steam production for this day (709,900 lbs) was lower than the average steam production of the whole month (884,465 lbs). A daily energy consumption calculation was necessary in order to estimate the energy consumption on March days for which steam consumption was not available. The energy consumption for March of 2015 was used to estimate the energy consumption of the months in which the temperature was cold (November – March), while the April of 2014 estimated energy consumption was used to estimate the energy consumption for the hot months (April – October). See Appendix H for detailed results of these estimates.

According to the estimated energy consumption calculations presented in Appendix H, it was found that there was significant energy consumption savings when using the Grundfos pumps as compared to the energy consumption of the Worthington pump. See Fig. 77 for the estimated monthly energy consumption of the Worthington pump and the Grundfos pumps.

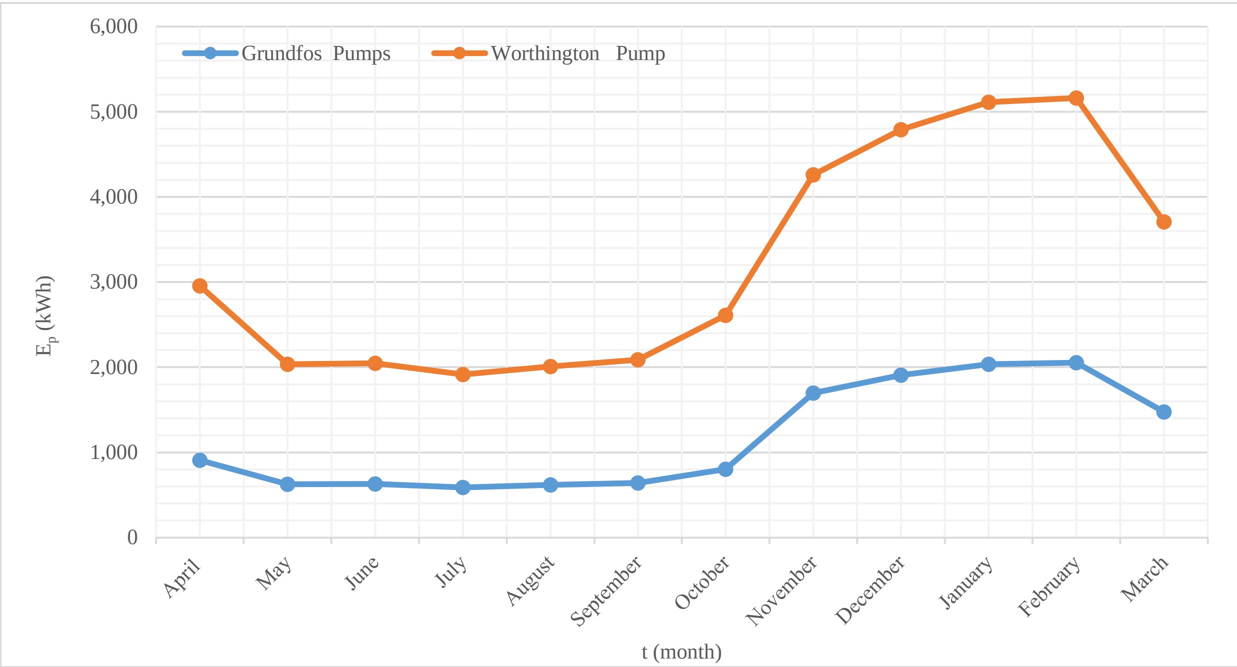


Figure 77: Monthly energy consumption of Worthington pump vs. Grundfos pumps

The annual energy consumption was 38,693.71 kWh (valued at $2,847.85 for 2014) for the Worthington pump while the annual energy consumption was 13,987.98 kWh (valued at $1,029.51 for 2014) for the Grundfos pumps. The projected 20 year present value of the energy consumption cost for this LCCA was $44,891 for the Worthington pump. On the other hand, the present value for 20 years of energy consumption cost was $16,228 for the Grundfos pumps. According to the LCCA, the present value of the 20 years of net savings from running the Grundfos pumps in level control mode was $28,663 in energy costs as compared to the Worthington pump (see Appendix I).

The present value of all LCCs for 20 years was $497,776 for the Worthington pump, and the present value of total LCC for the Grundfos pumps was $472,358. The water usage for both pumps was assumed to be the same for each summer and each winter of the 20 years. Based on the data available, it was assumed that the steam power plant consumed 2,449,270 gallons in the summer and 4,665,930 gallons in the winter. Therefore, the present value of the water consumption cost for 20 years was $412,870. Again, this value was the same for the Worthington pump and the Grundfos pumps because they were assumed to consume the same amount of water. The overall savings from running the Grundfos pumps in level control mode for 20 years, including all other costs, was $25,418. However, Case 2 did not consider the vent condenser energy impact that will be presented in next section. See Appendix I for the LCC reports.

4.2 Vent Condenser LCC

The previous section computed savings when both pump types were not providing water to the vent condenser. Therefore, the opportunity to reclaim the energy in the escaping non-condensable gases was not available.

Based on the energy calculations in Appendix D, the steam power plant would save 8,332,024,923 BTU (equivalent to 8,179,879.17 ft^3 of natural gas) from having the vent condenser running for one year. That tremendous amount of energy would be released to the atmosphere if the pumps were not providing water to the vent condenser. Based on that amount of estimated annual energy lost, the 20 year present value of the energy lost was $846,925. Therefore, after taking all of the 20 year costs that the steam power plant would have to pay for maintenance and purchasing the vent condenser, the vent condenser would save the steam power plant $812,574. One might notice that the savings from using the vent condenser far surpasses the savings from having the Grundfos pumps running in level control mode. That level of savings from the vent condenser would leave no option to have the vent condenser turned off. See Appendix J for the vent condenser's LCCA.

Chapter 5

Summary

- The Grundfos pumps were configured to provide water to the DA tank in level control mode without having problems of back flow and steam hammering. The minimum performance of the Grundfos pumps was carefully adjusted in order to have the pumps run at 42% of their full speed in order to meet the DA tank's demand flow rate. The pump controller's integral time (2 sec) was selected in order to run the pumps with a moderate sensitivity so as not to overshoot the water level in the DA tank.
- Pressure drop calculations were performed using all pipe friction equations in order to estimate the required pressure at point A in order to have the water lifted to the vent condenser (see Fig. 45); and the pressure at point A (see Fig. 45) was found to be 28 PSIG at a total Worthington pump flow rate of 214.7 GPM and a discharge pressure of 36.56 PSIG.
- Two case studies were presented. In Case 1, both the Grundfos and the Worthington pumps were running to provide the condensate water to the DA tank and the vent condenser. Case 2 was devised to have the two pump types running to provide the water to the DA tank only. The recorded power consumption showed that the Worthington pump was consuming more power (2.398 kW in Case 2, Test #1) than the Grundfos pumps, which were running in level control mode.
- LCCA was performed for Case 2 in order to determine the energy savings when running the Grundfos pumps in level control mode without having the control valve regulate the flow. The Grundfos pumps' savings was $25,418 for 20 years.

- ❖ LCCA was performed for the vent condenser in order to calculate the reclaimed energy savings that it provides for the steam power plant. The vent condenser savings over-shadowed the savings of the Grundfos pumps when running in level control mode. The vent condenser savings was $812,574 for 20 year.

Conclusions

In Case 2, the Grundfos pumps showed a significant energy consumption savings (running in level control mode to provide condensate water to the DA tank only). On the other hand, the Worthington pump was unable to save a noticeable amount of energy even when it only provided condensate water to the DA tank. Therefore, the Grundfos pumps were more beneficial in saving energy when the demand flow rate and pressure discharge head decreased. The total 20 years energy costs for the Grundfos pumps running in level control mode was $16,228, and, for the same task, the Worthington pump's 20 year energy cost was $44,891. The total 20 year LCC for the Worthington pump and the Grundfos pumps was $497,776 and $472,358, respectively. Therefore, the Grundfos CRE 15-3 was able to save $25,418 in 20 years. This savings is relatively low for a life span of 20 years.

The Grundfos pumps were only able to provide the water to the DA tank in the steam power plant when they ran in level control mode, and the vent condenser was isolated from the system. That caused the steam power plant to be unable to reclaim the energy escaping with the non-condensable gases. This caused the steam power plant to lose the vent condenser's energy savings of $812,574 in 20 years.

When running in level control mode, the Grundfos pumps' reaction to a water level change is not appropriate. That reaction causes the pumps increase their speed up to 100% of full speed.

Therefore, the pumps consume too much energy. In addition, the pumps supply a high flow rate to the DA tank, which requires a fairly steady flow rate so that the bled steam has the time to heat and dearate the condensate water.

The energy consumption of Worthington pump was found to be less than the energy consumption of Grundfos pumps in Case 1, Test #1. The Grundfos pumps consumed 8.433 kW while the Worthington pump consumed 5.567 kW, even though they were doing the same task (providing the condensate water to the DA tank and the vent condenser). However, the Grundfos pumps provided a relatively higher discharge pressure than the Worthington pump. The Grundfos pumps were set to provide constant discharge pressure of 43 PSIG while the Worthington pump could run at varied discharge pressures (see Fig. 49). Thus, the Grundfos pumps consumed more energy than the Worthington pump.

Again, the DA tank requires a steady flow rate so that the bled steam from the boilers has the time to heat and dearate the condensate water. The Worthington pump with a control valve regulator provides a smooth flow rate as compared to the Grundfos pumps' flow rate when running in level control mode (see Figs. 63 and 64). Thus, the Worthington pump is preferable for KU's steam power plant applications over the Grundfos pumps [when they run in level control mode].

Recommendations for Future Work

- ❖ This study shows appreciable energy savings from the vent condenser. Therefore, installing a new vent condenser next to the boiler blowdown heat recovery system is beneficial in order to help increase the make-up water temperature. Hence, the water temperature in the storage tanks will increase.

- Another vent condenser can be installed next to the one on the first floor in order to capture more energy from the non-condensable gases.
- Change the Grundfos pumps' minimum performance to the situation wherein the two pumps run at 42% of their full speed so that more consistent flow can reach the DA tank. Therefore, there should be no sudden drop in the water level in the DA tank. However, one must be careful not to over fill the DA tank when the steam demand decreases.
- Run the Grundfos pumps at a lower pressure in pressure control mode, especially in the summer. Therefore, the control valve should waste less energy under such operating conditions. In addition, energy savings can be achieved since pressure directly affects the energy consumption (see Eq. 18). This type of operation condition cannot be performed with the Worthington pump.
- The steam power plant's boilers run at a pressure of 170-175 PSIG. However, the boiler feed pumps provide pressures up to 350 PSIG. Therefore, another energy savings potential can be investigated by replacing the existing constant speed boiler pumps with variable speed pumps. A simulation study can be performed in order to investigate the energy savings from replacing the constant speed boiler feed pumps with variable speed pumps.
- In order to have the Grundfos pumps be able to feed the vent condenser when running in level control mode, the best option would be to have the vent condenser installed in the basement. Consequently, the extra 28 PSIG of discharge pressure would not be necessary to lift the water to the current vent condenser location (on the first floor).
- The steam power plant could have a pump installed at point A (see Fig. 45) with a rated efficiency of 70% and provide a flow rate of 82 GPM. Using Eq. (18) to calculate the required power to run this pump, this pump would consume another 1.913 HP (1.426 kW).

Hence, more power would be consumed in addition to the Grundfos pumps running in level control mode (1.592 kW) as well as the extra pump's maintenance. Therefore, from the energy consumption point-of-view, the total energy consumption of the two pumps would be 3 kW. However, even though the total energy consumption would increase, the suggested pump would help to lift condensate water to the vent condenser to reclaim the energy lost from the non-condensable gases. The two pumps would consume 24.8% less energy (0.99 kW) than the Worthington pump (3.99 kW).

- When the Grundfos pumps run in level control mode, another flow meter could be installed that measures the exit flow from the DA tank and signals the Grundfos pumps to supply the same flow rate that exits from the DA tank. Thus, with this configuration the pumps would not provide a wave-like flow rate to the DA tank.

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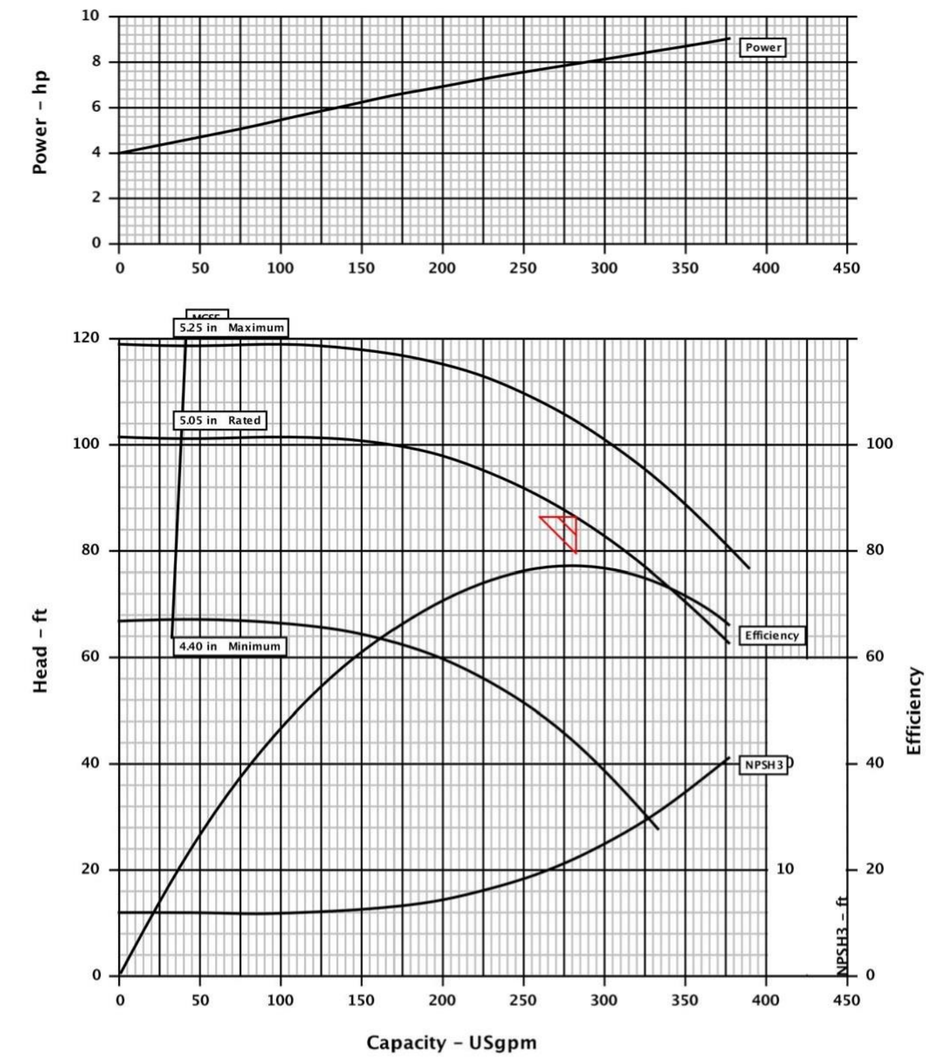
Appendix A: All Pumps Curves and Specifications

A1. Worthington D-824 Constant Speed Pump Curves

FLOWSERVE

		Pump size & type	: D824-3X2X5F
		Based on curve no.	: A-19446
		Number of stages	: 1
Customer	:	Capacity	: 282.9 USgpm
Item number	: -	Head	: 86.44 ft
Service	:	Specific gravity	: 1.000
Flowserve reference	: 363650489	Pump speed	: 3550 rpm
Date	: April 8, 2015	Test tolerance	: Hydraulic Institute Level B

CURVES ARE APPROXIMATE, PUMP IS GUARANTEED FOR ONE SET OF CONDITIONS; CAPACITY, HEAD, AND EFFICIENCY.



Affinity v1.3.5
1 of 3

FLOWSERVE

Hydraulic Datasheet

Customer	:	Pump / Stages	: D824-3X2X5F / 1
Customer reference	: Douglas Pump	Based on curve no.	: A-19446
Item number	: -	Flowserve reference	: 363650489
Service	:	Date	: April 8, 2015

Operating Conditions

Capacity	: 282.9 USgpm
Water capacity (CQ=1.00)	: -
Normal capacity	: -
Total developed head	: 86.44 ft
Water head (CH=1.00)	: -
NPSH available (NPSHa)	: Ample
NPSHa less NPSH margin	: -
Maximum suction pressure	: 0.0 psig

Liquid

Liquid type	: Other
Temperature / Spec. Gravity	: 60 F / 1.000
Solid Size - Actual / Limit	: - / 0.50 in
Viscosity / Vapor pressure	: 1.00 cSt / -

Materials / Specification

Material column code	: STD
Pump specification	: General Industrial

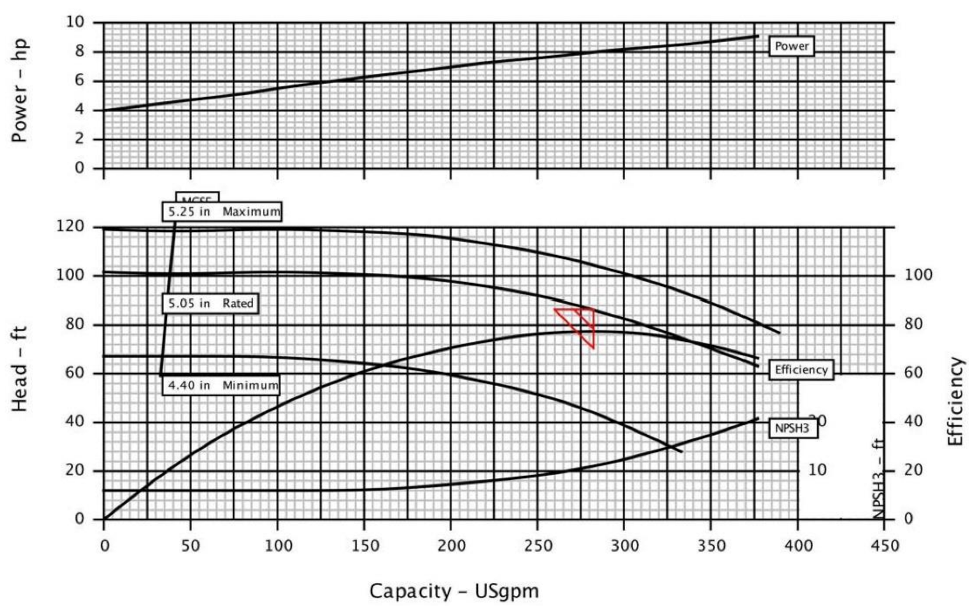
Other Requirements

Hydraulic selection : No specification
Construction : No specification
Test tolerance : Hydraulic Institute Level B
Driver Sizing : Max Power(MCSF to EOC)with SF

Performance

Hydraulic power	: 6.18 hp	Impeller diameter	
Pump speed	: 3550 rpm	Rated	: 5.05 in
Pump overall efficiency (CE=1.00)	: 77.5 %	Maximum	: 5.25 in
		Minimum	: 4.40 in
NPSH required (NPSH3)	: 11.1 ft	Suction specific speed	: 9190 US units
Rated power	: 7.97 hp	Minimum continuous flow	: 38.5 USgpm
Maximum power	: 9.04 hp	Maximum head @ rated dia	: 101.36 ft
Driver power	: 10.00 hp / 7.46 kW	Flow at BEP	: 282.9 USgpm
Casing working pressure	: 43.9 psig	Flow as % of BEP	: 100.0 %
(based on shut off @ cut dia/rated SG)		Efficiency at normal flow	: -
Maximum allowable	: 175.0 psig	Impeller dia ratio (rated/max)	: 96.2 %
Hydrostatic test pressure	: 265.0 psig	Head rise to shut off	: 17.3 %
Est. rated seal chamb. press.	: -	Total head ratio (rated/max)	: 82.6 %

CURVES ARE APPROXIMATE, PUMP IS GUARANTEED FOR ONE SET OF CONDITIONS; CAPACITY, HEAD, AND EFFICIENCY.



Affinity v1.3.5

FLOWSERVE

Construction Datasheet

Customer	:	Pump / Stages	: D824-3X2X5F / 1
Customer reference	: Douglas Pump	Based on curve no.	: A-19446
Item number	: -	Flowserve reference	: 363650489
Service	:	Date	: April 8, 2015

Construction

Nozzles	Size	Rating	Face	Pos'n
Suction	3.00	125#	FF	End
Discharge	2.00	125#	FF	Top

Casing mounting	: Foot
Casing split	: Radial
Impeller type	: Closed Impeller
Bearing type (radial)	: N/A
Bearing number (radial)	: N/A
Bearing type (thrust)	: N/A
Bearing number (thrust)	: N/A
Bearing lubrication	: Other
Rotation (view from cplg)	: CW per Hyd. Institute

Materials

Casing	: Cast Iron 25
Impeller	: Bronze
Case wear ring	: N/A
Impeller wear ring	: N/A
Inducer	: N/A
Shaft	: Steel
Sleeve	: Bronze

Baseplate, Coupling and Guard

Baseplate type	: N/A
Baseplate material	: N/A
Coupling manufacturer	: N/A
Coupling size	:
Coupling / Shaft guard	: N/A

Weights (Approx.)

Bareshaft pump(nett)	: 58.0 lb
Baseplate(nett)	: -
Driver(nett)	: 127.0 lb
Shipping gross weight/vol.	: 212.7 lb / 6199 cu.in

Testing

Hydrostatic test	: None
Performance test	: None
NPSH test	: None

Paint and Package

Pump paint	: FPD Standard
Base grout surface prep	: FPD Std.
Shipment type	: Domestic

Driver Information

Manufacturer	: Flowserve Choice
Power	: 10.00 hp / 7.46 kW
Service factor (req'st / act)	: 1.0 / -
Speed	: 3600
Orientation / Mounting	: Horizontal / Foot and flange
Driver Type	: NEMA
Frame-size / material	: 215JM / Aluminum
Enclosure	: TEFC
Hazardous area class	: -
Explosion 'T' rating	: -
Volts / Phase / Hz	: 230/460 / 3 / 60 Hz
Amps-full load/locked rotor	: 12.00 A / 72.10 A
Motor starting	: Direct on line (DOL)
Insulation	: F
Temperature rise	: 80 C
Bearings	: Ball
Lubrication	: Grease
Motor mounted by	: Flowserve

Sound Pressure (dBA @ 1.0 m)

Driver, expected	: -
Pump & driver, estimated	: -

Seal Information

Arrangement	: Sgl. Int. O-Ring
Size	: 1.375
Manufacturer / Type	: Flowserve / PAC 51
Material code (Man'f/API)	: BCFXF / -
Internal neck bushing	: N/A

Gland

Gland material	: N/A
Flush	: N/A
Vent	: N/A
Drain	: N/A
Auxiliary seal device	: N/A

Piping

Seal flush plan	: None
Seal flush construction	: -
Seal flush material	: -
Aux seal flush plan	: None
Aux seal flush construction	: -
Aux seal flush material	: -

Notes

-

-

-

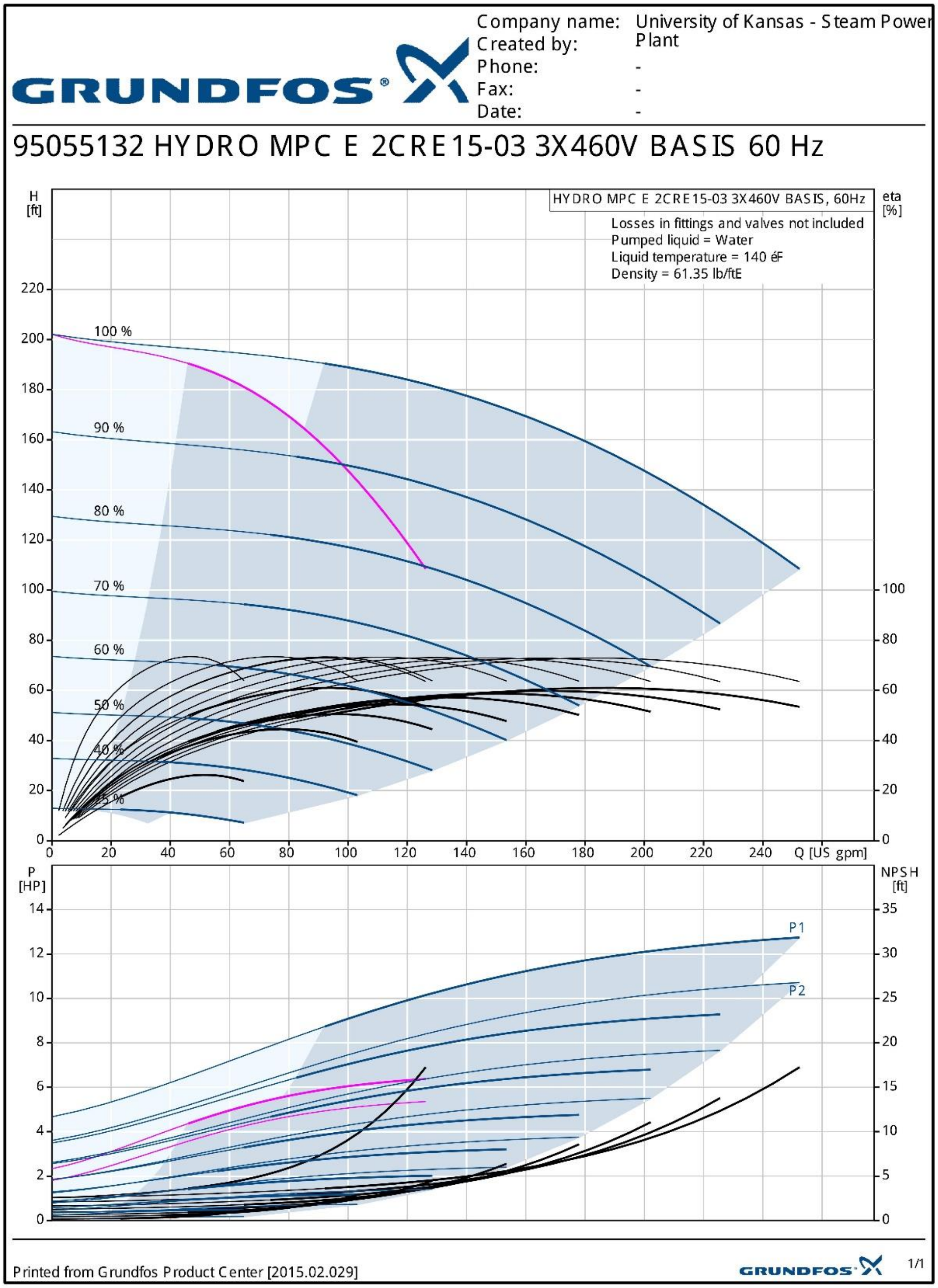
-

Bronze Adapter Wear Ring

-

Affinity v1.3.5

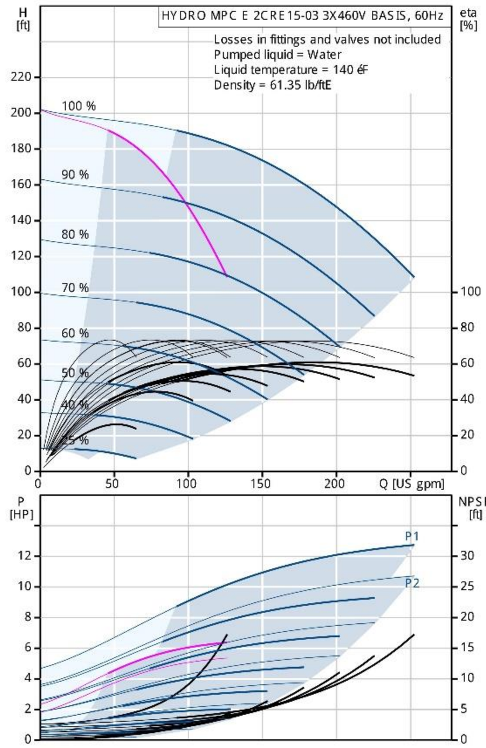
A2. Grundfos CRE 15-3 Variable Speed Pumps Curves



Company name: University of Kansas - Steam Power Plant
Created by:
Phone: -
Fax: -
Date: -

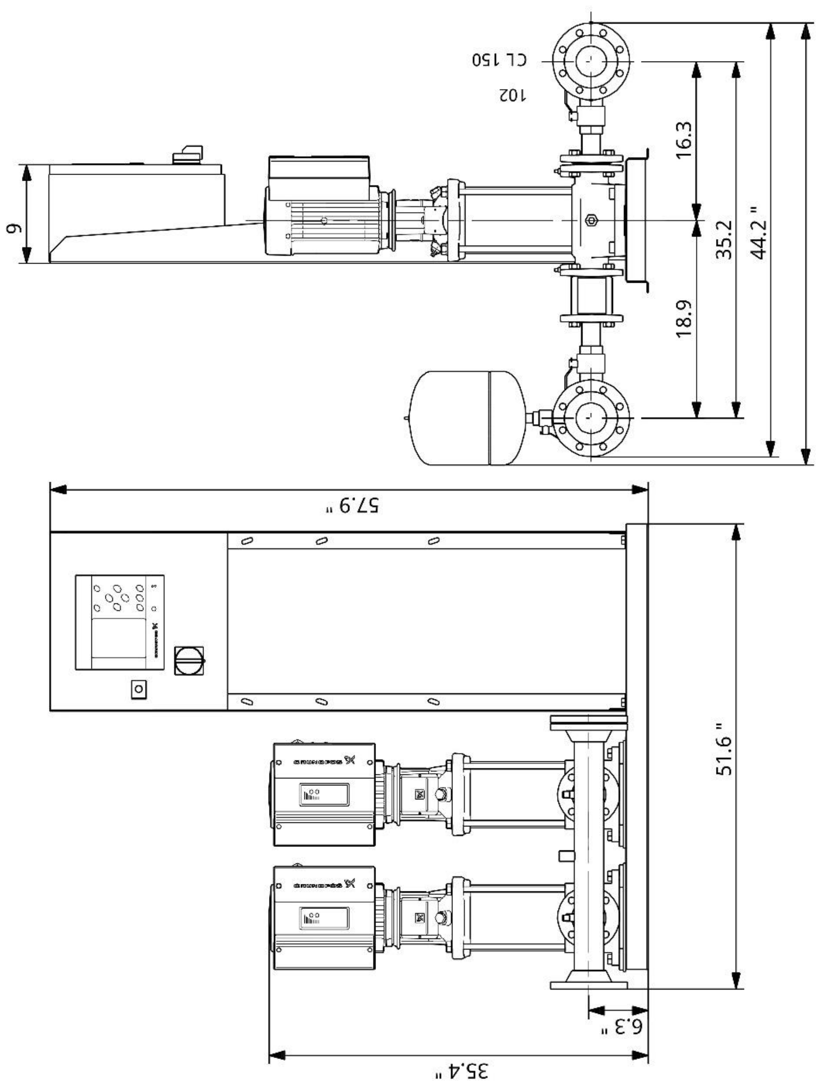
GRUNDFOS

Description	Value
General information:	
Product name:	HYDRO MPC E 2CRE15-03 3X460V BASIS
Position	
Product No.:	95055132
EAN:	5700836977051
Price:	On request
Technical:	
Max flow:	244 US gpm
Max flow system:	244 US gpm
Head max:	200 ft
Impellers main:	3
Main pump name:	CRE15-03
Main pump Number:	96541270
Number of pumps:	2
Non-ret. valve:	at discharge side
Installation:	
Maximum operating pressure:	232 psi
Maximum inlet pressure:	145 psi
Flange standard:	ANSI
Pump inlet:	102
Pump outlet:	102
Pressure stage:	CL 150
Liquid:	
Liquid temperature range:	32 .. 140 éF
Electrical data:	
Power (P2) main pump:	7.51 HP
Main frequency:	60 Hz
Rated voltage:	3 x 3X460-480V, 60 Hz
Starting main:	electronically
Rated current of system:	20.8 A
Enclosure class (IEC 34-5):	UL Type 3R/12
Controls:	
Control type:	E
Operation unit:	CU 352
Tank:	
Diaphragm tank:	No
Others:	
Net weight:	650 lb
Language:	EN
Product range:	NAMREG
Configuration file Hydro MPC:	95043547



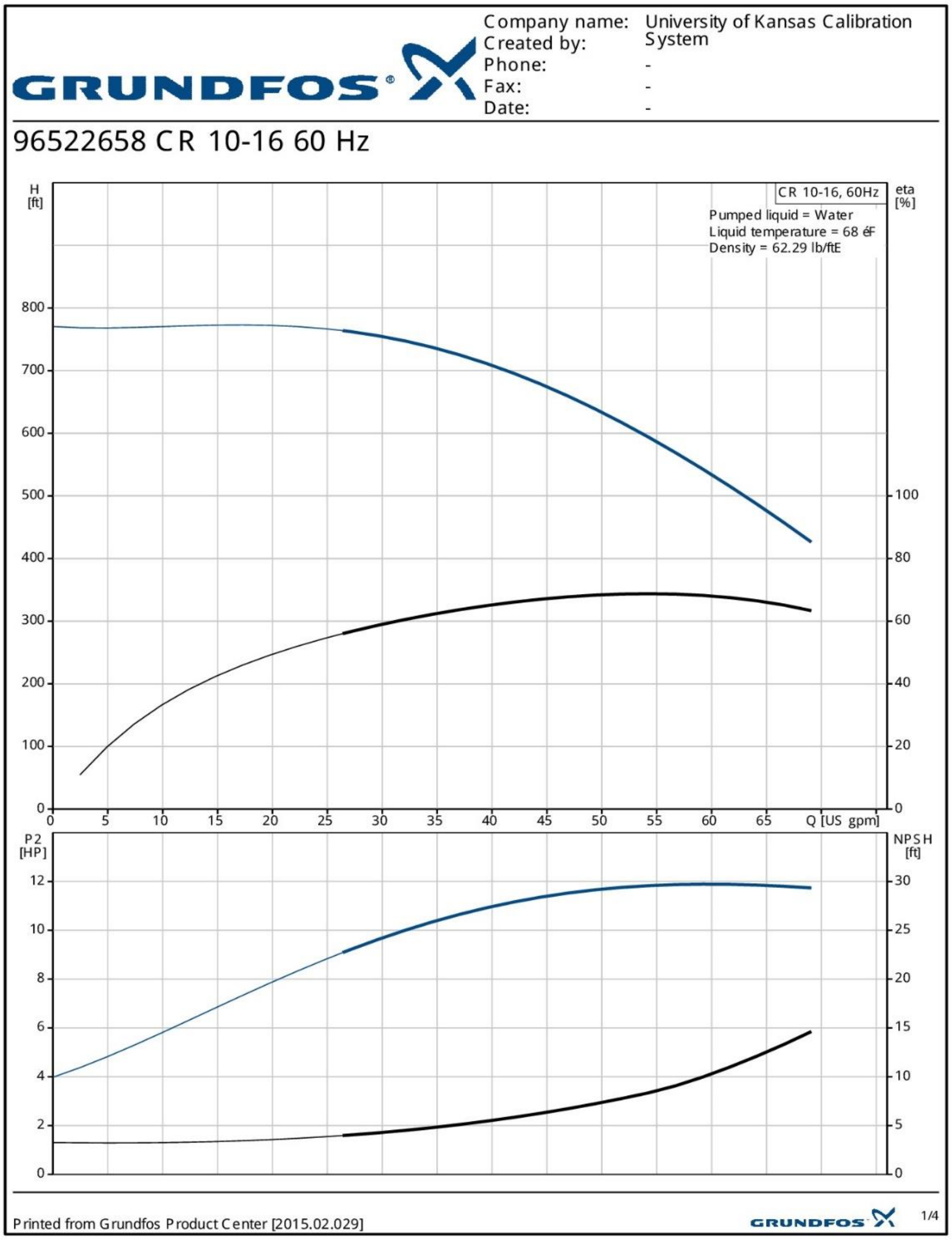
Company name:	University of Kansas - Steam Power Plant
Created by:	
Phone:	-
Fax:	-
Date:	-

95055132 HYDRO MPC E 2CRE15-03 3X460V BASIS 60 Hz



Note! All units are in [mm] unless otherwise stated.
Disclaimer: This simplified dimensional drawing does not show all details.

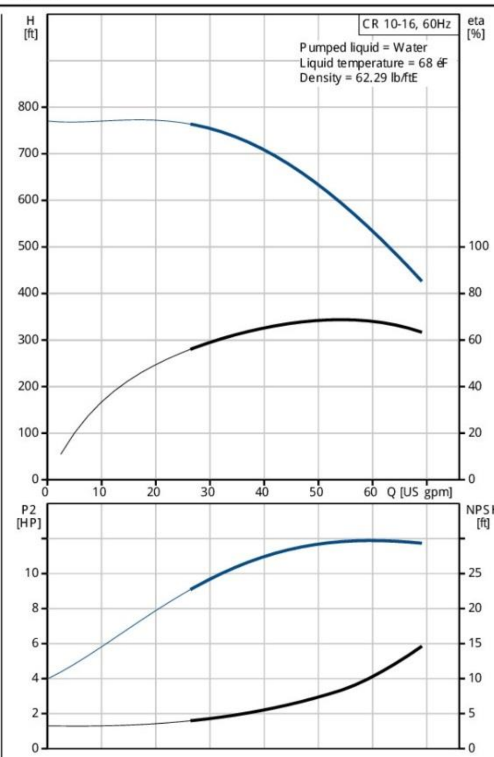
A3. CR10-16 Pump Curves and Specifications



GRUNDFOS

Company name:	University of Kansas Calibration System
Created by:	
Phone:	-
Fax:	-
Date:	-

Description	Value
General information:	
Product name:	CR 10-16 A-GJ-A-E-HQQE
Position	
Product No.:	96522658
EAN:	5700396891217
Price:	On request
Technical:	
Speed for pump data:	3444 rpm
Rated flow:	53.3 US gpm
Rated head:	604 ft
Head max:	771 ft
Impellers:	16
Shaft seal:	HQQE
Approvals on nameplate:	ANSI/NSF61
Curve tolerance:	ISO9906:2012 3B
Stages:	16
Pump version:	A
Model:	A
Cooling:	ODP
Materials:	
Pump housing:	Cast iron
	EN-JL1030
	ASTM A48-30 B
Impeller:	Stainless steel
	DIN W.-Nr. 1.4301
	AISI 304
Material code:	A
Code for rubber:	E
Installation:	
Maximum ambient temperature:	104 éF
Max pressure at stated temperature:	363 psi / 250 éF
	363 psi / -4 éF
Flange standard:	ANSI
Connect code:	GJ
Pipe connection:	2"
Pressure stage:	Class 250
Flange size for motor:	254TC
Liquid:	
Liquid temperature range:	-4 .. 248 éF
Electrical data:	
Motor type:	BALDOR
Number of poles:	2
Rated power - P2:	15 HP
Power (P2) required by pump:	15 HP
Main frequency:	60 Hz
Rated voltage:	3 x 208-230/460 V
Service factor:	1,15
Rated current:	37-35/17.5 A
Rated speed:	3450 rpm
Insulation class (IEC 85):	F
Motor protection:	NONE
Motor Number:	84Z05097

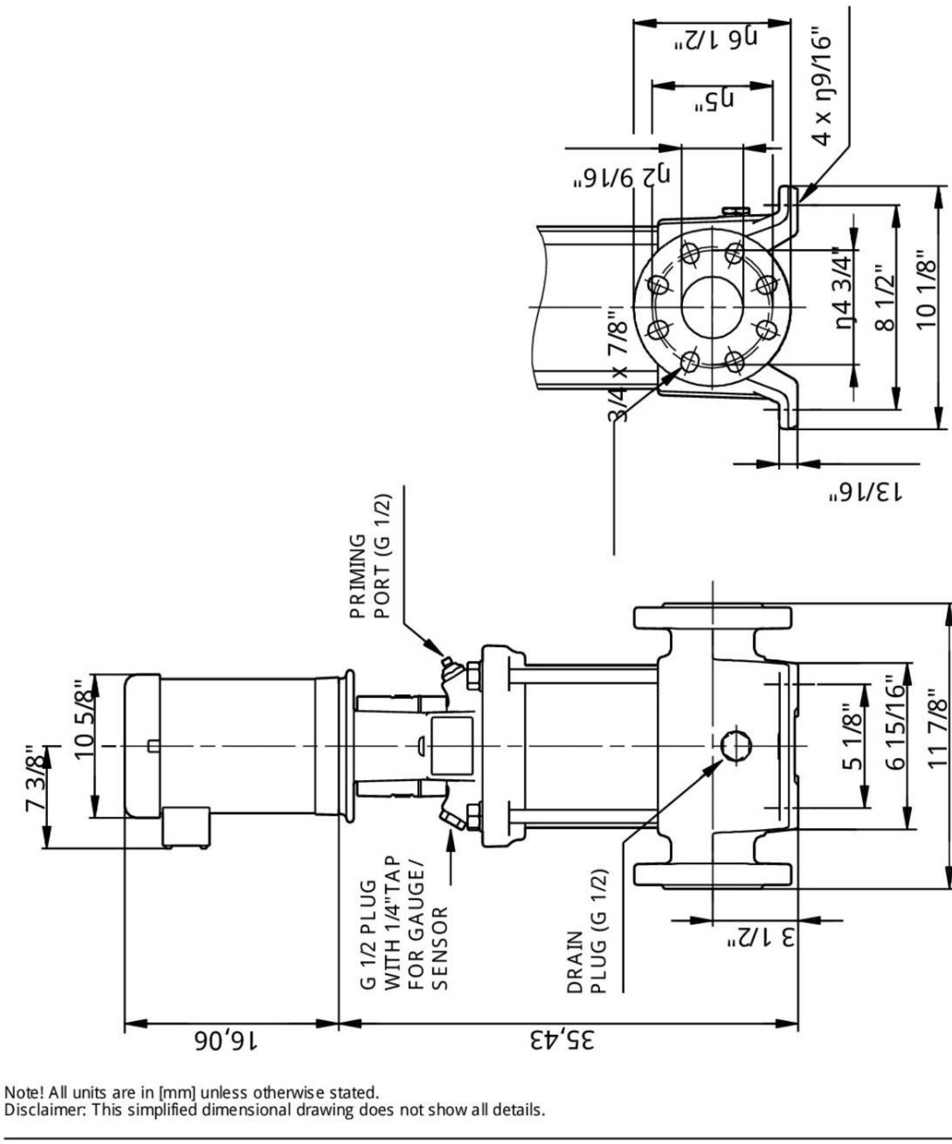


Description	Value
Others:	
Net weight:	265 lb
Gross weight:	377 lb
Shipping volume:	12 ftE

GRUNDFOS

Company name: University of Kansas Calibration System
Created by:
Phone: -
Fax: -
Date: -

96522658 CR 10-16 60 Hz



Note! All units are in [mm] unless otherwise stated.
Disclaimer: This simplified dimensional drawing does not show all details.

Appendix B: All Measuring Devices Used in the Project

B1. Mastech HY3003D Power Supply

Power Supply HY30XX Series

MODEL EXPLAINATION:
HY XX XX X - X
① ② ③ ④ ⑤

① Products of MASTEC
② Output voltage numbers
③ Output current numbers
④ no: LED display
D: LCD display
C: two pointer meters display
S: four pointer meters display
⑤ no: single output voltage current regulated
2: double output voltage current regulated
3: double output voltage current regulated + fixed 5V3A

MODEM	HY3002D	HY3003D	HY3005D
Input Voltage	110/220V±10%		
Output Voltage	0 ~ 30V		
Output Current	0 ~2A	0 ~ 3A	0 ~ 5A
Source Effect	CV ≤ 0.01%±1mV; CC ≤ 0.02%±1mA		
Load Effect	CV ≤ 0.01%±5mV; CC ≤ 0.02%±5mA		
Ripple & Noise	≤ 1mVrms		
Display	Two 3 1/2 digit LCD display		
Accuracy	V: ±1%±2digits; C: ±2%±2digits		
Size	291 x 158 x 136mm		
Weight	3 ~ 6kg		

MODEM	HY3002C	HY3003C	HY3005C
Input Voltage	110/220V ± 10%AC		
Output Voltage	0 ~ 30V		
Output Current	0 ~ 2A	0 ~ 3A	0 ~ 5A
Source Effect	CV≤ 0.01%±1mV; CC≤ 0.02%±1mA		
Load Effect	CV≤ 0.01%±5mV; CC≤ 0.02%±5mA		
Ripple & Noise	≤ 1mVrms		
Display	Voltage & Amperometer display		
Accuracy	2.5%f.s.		
Size	291 x 158 x 136mm		
Weight	3 ~ 6kg		

Sinotech Shanghai
Room 1404, 1759 North Zhongshan Road, Putuo District,
Shanghai, China

Page 1

B2. HOBO H12-006 Data Logger

onset
HOBO® Data Loggers

HOBO® U12 Logger

Multi-channel energy & environmental monitoring

HOBO U12 data loggers provide flexibility for monitoring up to 4 channels of energy and environmental data with a single, compact logger. They provide 12-bit resolution measurements for detecting greater variability in recorded data, direct USB connectivity for convenient, fast data offload, and a 43K measurement capacity.

Supported Measurements: Temperature, Relative Humidity, Dew Point, 4-20mA, AC Current, AC Voltage, Air Velocity, Carbon Dioxide, Compressed Air Flow, DC Current, DC Voltage, Gauge Pressure, Kilowatts, Light Intensity, Volatile Organic Compound (some sensors sold separately)

Key Advantages:

- Records up to 4 channels
- Your choice of three models, with flexible measurement options
- Programmable as well as push-button start
- Compatible with a broad range of external sensors

Minimum System Requirements:

Software USB cable*

For complete information and accessories, please visit: **www.onsetcomp.com**

Part number	U12-006 (4 Ext)	U12-012 (Temp/RH/Light/Ext)	U12-013 (Temp/RH/2 Ext)
Memory	43,000 measurements		
Sampling rate	1 second to 18 hours, user-selectable		
Battery life	1 year typical, user-replaceable, CR2032		
		Temperature	
Max range		-20° to 70°C (-4° to 158°F)	
Accuracy		± 0.35°C from 0° to 50°C (± 0.63°F from 32° to 122°F)	
Resolution (12-bit)		0.03°C @ 25°C (0.05°F @ 77°F)	
		Relative Humidity	
Measurement range		5% to 95% RH (non-condensing)	
Accuracy		± 2.5% typical, 3.5% maximum, from 10 to 90% RH	
Resolution (10-bit)		0.03% RH	
		Light Intensity	
		Designed for general purpose indoor measurement of relative light levels	
Range		1 to 3000 footcandles (lumens/ft2) typical 0-32,300 lumens/m2	
	External Input		
Range	0 to 2.5 VDC		
Accuracy	± 2 mV, ± 2.5% of absolute reading		
Resolution	0.6 mV		
CE compliant	Yes		

*USB cable included with software
For stand-alone data logging applications in harsh indoor environments, see the 4-channel HOBO U12 Industrial data logger (U12-008) at onsetcomp.com

Contact Us

Sales (8am to 5pm ET, Monday through Friday)
- Email sales@onsetcomp.com
- Call 1-800-564-4377
- Fax 508-759-9100

Technical Support
- (8am to 8pm ET, Monday through Friday)
- Contact Product Support
- Call 877-564-4377

B3. Danfoss Pressure Transducer

Danfoss

Pressure transmitter for industrial applications
Type MBS 3000

Technical brochure

Features

- Designed for use in severe industrial environments
- Enslosure and wetted parts of acid-resistant stainless steel (AISI 316L)
- Pressure ranges in relative (gauge) or absolute from 0 up to 600 bar
- All standard output signals: 4 - 20 mA, 0 - 5 V, 1 - 5 V, 1 - 6 V, 0 - 10 V, 1 - 10 V
- A wide range of pressure and electrical connections
- Temperature compensated and laser calibrated

Description

The compact pressure transmitter MBS 3000 is designed for use in almost all industrial applications, and offers a reliable pressure measurement, even under harsh environmental conditions.

The flexible pressure transmitter programme covers different output signals, absolute and gauge (relative) versions, measuring ranges from 0-1 to 0-600 bar and a wide range of pressure and electrical connections.

Excellent vibration stability, robust construction, and a high degree of EMC/EMI protection equip the pressure transmitter to meet the most stringent industrial requirements.

Ordering standard versions

Plug: Pg 9 (EN 175301-803-A)
Output signal: 4-20 mA
Pressure connection:
G ¼ A (EN 837)

Measuring range P_e[1) [bar]	Type	Code no.
0 - 1	MBS 3000 - 1011 - 1 AB04	**060G1113**
0 - 1.6	MBS 3000 - 1211 - 1 AB04	**060G1429**
0 - 2.5	MBS 3000 - 1411 - 1 AB04	**060G1122**
0 - 4	MBS 3000 - 1611 - 1 AB04	**060G1123**
0 - 6	MBS 3000 - 1811 - 1 AB04	**060G1124**
0 - 10	MBS 3000 - 2011 - 1 AB04	**060G1125**
0 - 16	MBS 3000 - 2211 - 1 AB04	**060G1133**
0 - 25	MBS 3000 - 2411 - 1 AB04	**060G1430**
0 - 40	MBS 3000 - 2611 - 1 AB04	**060G1105**
0 - 60	MBS 3000 - 2811 - 1 AB04	**060G1106**
0 - 100	MBS 3000 - 3011 - 1 AB04	**060G1107**
0 - 160	MBS 3000 - 3211 - 1 AB04	**060G1112**
0 - 250	MBS 3000 - 3411 - 1 AB04	**060G1111**
0 - 400	MBS 3000 - 3611 - 1 AB04	**060G1109**
0 - 600	MBS 3000 - 3811 - 1 AB04	**060G1110**

[1) *Relative/ gauge*

Ordering of special versions

MBS 3000 - [ ][ ][ ][ ] - [ ][ ][ ][ ]

Preferred versions (shown shaded)

Non-standard build-up combinations may be selected. However, minimum order quantities may apply. Please contact your local Danfoss office for further information or request for other versions.

Measuring range

Range	Code
0 - 1 bar	1 0
0 - 1.6 bar	1 2
0 - 2.5 bar	1 4
0 - 4 bar	1 6
0 - 6 bar	1 8
0 - 10 bar	2 0
0 - 16 bar	2 2
0 - 25 bar	2 4
0 - 40 bar	2 6
0 - 60 bar	2 8
0 - 100 bar	3 0
0 - 160 bar	3 2
0 - 250 bar	3 4
0 - 400 bar	3 6
0 - 600 bar	3 8

Pressure reference

Reference	Code
Gauge (relative)	1
Absolute	2

Pressure connection

Code	Connection
A B 0 4	G ¼ A (EN 837)
A B 0 6	G $^3/_8$ A (EN 837)
A B 0 8	G ½ A (EN 837)
A C 0 4	¼ -18 NPT
A C 0 8	½ -14 NPT
G B 0 4	DIN 3852-E-G ¼ Gasket: DIN 3869-14 NBR

Electrical connection

Figures refer to plug and standard PIN configuration - see page 5

Code	Connection
1	Plug Pg 9 (EN 175301-803-A)
2	*)Plug, AMP Econoseal, J series, male, excl. female plug
3	Screened cable, 2 m
5	*)Plug, IEC 947-5-2, M12 × 1, male, excl. female plug
8	*) Plug, AMP Superseal 1.5 series male, excl. female plug

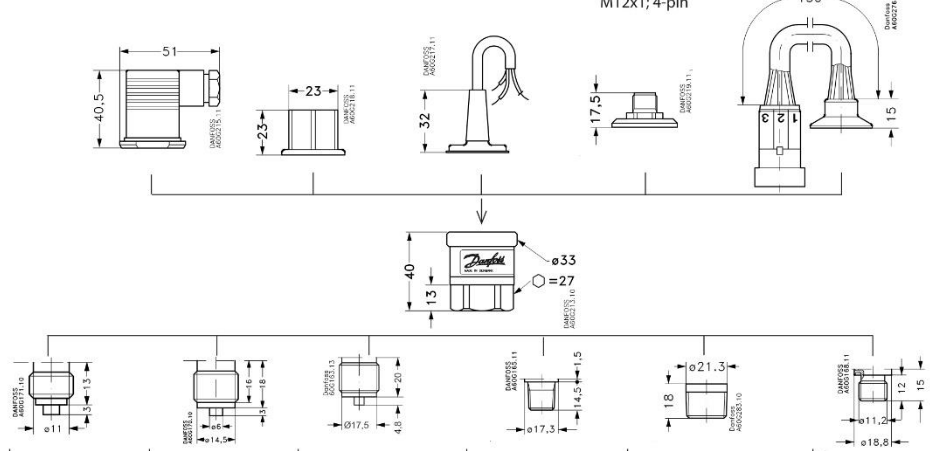
Output signal

Code	Signal
1	4 - 20 mA
2	0 - 5 V
3	1 - 5 V
4	1 - 6 V
5	0 - 10 V
7	1 - 10 V

*) Gauge versions only available as sealed gauge versions

Dimensions / Combinations

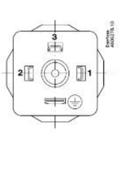
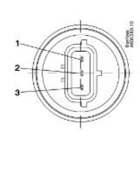
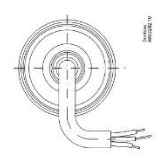
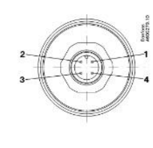
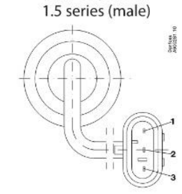
Type code	1	2	3	5	8
	EN 175301-803-A, Pg 9	AMP Econoseal	2 m screened cable	EN 60947 - 5 - 2 M12x1; 4-pin	AMP Superseal



	G¼ A (EN 837)	G $^3/_8$ A (EN 837)	G ½ A (EN 837)	¼ - 18 NPT	½ - 14 NPT	DIN 3852-E-G ¼ Gasket: DIN 3869-14
Type code	AB04	AB06	AB08	AC04	AC08	GB04
Recommended torque 1)	30-35 Nm	30-35 Nm	30-35 Nm	2-3 turns after finger tightened	2-3 turns after finger tightened	30-35 Nm

1)Depends of different parameters as packing material, mating material, thread lubrication and pressure level.

Electrical connections

Type code, page 4				
1	2	3	5	8
EN 175301-803-A, Pg 9	AMP Econoseal J series (male)	2 m screened cable	EN 60497-5-2 M12x1 4-pin	AMP Superseal 1.5 series (male)
				
Ambient temperature				
–40 → + 85 °C	–40 → + 85 °C	–30 → +85 °C	–25 → +85 °C	–40 → +85 °C
Enclosure (IP protection fulfilled together with mating connector)				
IP 65	IP 67	IP 67	IP 67	IP 67
Materials				
Glass filled polyamid, PA 6.6	Glass filled polyamid, PA 6.6[1]	Poliolyfin cable with PE shrinkage tubing	Nickel plated brass, CuZn/Ni	Glass filled polyamid, PA 6.6[2]
Electrical connection, 4 - 20 mA output (2 wire)				
Pin1: + supply Pin 2: ÷ supply Pin 3: Not used Earth: Connected to MBS enclosure	Pin 1: + supply Pin 2: ÷ supply Pin 3: not used	Brown wire: + supply Black wire: ÷ supply Red wire: Not used Orange: Not used Screen: Not connected to MBS enclosure	Pin 1: + supply Pin 2: Not used Pin 3: Not used Pin 4: ÷ supply	Pin 1: + supply Pin 2: ÷ supply Pin 3 Not used
Electrical connection, 0 - 5V, 1 - 5 V, 1 - 6 V, 0 - 10 V, 1 - 10 V output				
Pin 1: + supply Pin 2: ÷ supply Pin 3: Output Earth: Connected to MBS enclosure	Pin 1: + supply Pin 2: ÷ supply Pin 3: Output	Brown wire: Output Black wire: ÷ supply Red wire: + supply Orange: Not used Screen: Not connected to MBS enclosure	Pin 1: + supply Pin 2: Not used Pin 3: Output Pin 4: ÷ supply	Pin 1: + supply Pin 2: ÷ supply Pin 3: Output

[1] Female plug: Glass filled polyester, PBT
[2] Wire: PETFE (teflon)
Protection sleeve: PBT mesh (polyester)

B4. Omega Pressure Sensor (PX43E0-200GI)

HEAVY-DUTY FLUSH DIAPHRAGM TRANSMITTER
½ NPT THREAD

CURRENT OUTPUT PRESSURE TRANSDUCERS

B

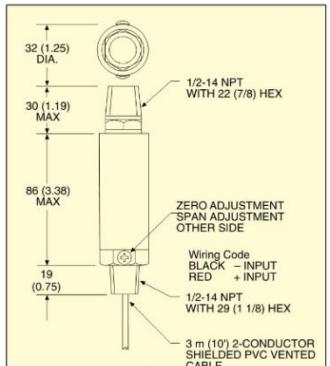
4 to 20 mA Output
0-50 to 0-750 psi
0-3 to 0-50 bar

1 bar = 14.5 psi
1 kg/cm² = 14.22 psi
1 atmosphere = 14.7 psi = 29.93 inHg = 760.2 mmHg = 1.014 bar

PX43E0-100GI, shown actual size.

PX43 Series

MADE IN USA · NIST Standard · CE · OMEGADYNE · RoHS



Dimensions: mm (inch)

- Stainless Steel Construction
- Processing or Industrial Applications
- Rugged Flush Diaphragm for Measurement of Difficult Fluids
- 3 m (10') Cable with Conduit Connection for Installation in Harsh Environments
- Heavy-Duty ½ NPT Fitting
- 4 to 20 mA Output for Noise-Free Transmission

OMEGA's PX43 is a high-accuracy, current output, industrial pressure transmitter with a heavy-duty flush diaphragm. It is designed for use with food and industrial fluids and slurries that are difficult to measure because of sticking or plugging of orifices. Its hermetically sealed, all stainless steel construction make it suitable for the harshest industrial environments. Ten feet of 2-conductor shielded cable is standard with a second ½ NPT fitting on the body for conduit installation. Pressure ranges from 0 to 50 up to 0 to 750 psi are available to cover most processing and industrial applications.

SPECIFICATIONS

Excitation: 10 to 40 Vdc
Output: 4 to 20 mA 10% adj
Zero Balance: 4 mA +10% -2% adj
Accuracy: 0.5% linearity, hysteresis and repeatability combined
Operating Temp Range: -46 to 121°C (-50 to 250°F)
Compensated Temp Range: 16 to 71°C (60 to 160°F)
Thermal Effects:
Span: 0.003% of rdg/°F
Zero: 0.0045% of FSO/°F
Proof Pressure: 150% of range
Burst Pressure: 300% of range
Wetted Parts: 17-4 PH stainless steel with 316 stainless steel diaphragm
Pressure Port: ½-14 NPT male
Electrical Connection: 3 m (10') 2-conductor shielded vented cable with ½-14 NPT fitting

To Order Visit omega.com/px43-i for Pricing and Details

RANGE		MODEL NO.	COMPATIBLE METERS*
0 to 50 psig	0 to 3.4 bar	**PX43E0-050GI**	DP41-E, DP25B-E, DP24-E
0 to 60 psig	0 to 4.1 bar	**PX43E0-060GI**	DP41-E, DP25B-E, DP24-E
0 to 100 psig	0 to 6.9 bar	**PX43E0-100GI**	DP41-E, DP25B-E, DP24-E
0 to 200 psig	0 to 13.8 bar	**PX43E0-200GI**	DP41-E, DP25B-E, DP24-E
0 to 300 psig	0 to 20.7 bar	**PX43E0-300GI**	DP41-E, DP25B-E, DP24-E
0 to 500 psig	0 to 34.5 bar	**PX43E0-500GI**	DP41-E, DP25B-E, DP24-E
0 to 750 psig	0 to 51.7 bar	**PX43E0-750GI**	DP41-E, DP25B-E, DP24-E

Comes complete with 5-point NIST traceable calibration.
Metric ranges available – consult Engineering.

* *See* **omega.com** *for compatible meters.*

Ordering Examples: PX43E0-100GI, 100 psi gage model with stainless steel wetted parts, 3 m (10') cable, 4 to 20 mA output. **PX43E0-050GI,** 50 psi gage model with stainless steel wetted parts, 3 m (10') cable, 4 to 20 mA output.

B-182

B5. Grundfos Differential Pressure Sensor

GRUNDFOS DATA SHEET

DPI 0 - 2.5

Differential Pressuresensor, Industry, 0 - 2.5 bar

TM04 4738 1909

Fig. 1 DPI sensor

Technical overview

Grundfos Direct Sensors ™ , type DPI, is a series of differential pressure sensors for industry. The DPI sensors are compatible with wet, aggressive media and are available for differential pressure ranges of 0 - 0.6 up to 0 - 10 bar.

The DPI sensor utilises MEMS sensing technology in combination with a novel packaging concept using corrosion-resistant coating on the MEMS sensing element. This makes the DPI sensor very robust and ideal for pump integration and monitoring in harsh environments.

Applications

- Pump and pump control systems
- Filters (monitoring)
- Cooling and temperature control systems
- Water treatment systems
- Boiler control systems
- Renewable energy systems
- Heat exchanger efficiency (monitoring of fouling).

Features

- Pressure ranges: 0 - 0.6; 0 - 1; 0 - 1.2; 0 - 1.6; 0 - 2.5; 0 - 4; 0 - 6 and 0 - 10 bar differential pressure
- Designed for harsh environments
- Analogue output signal
- Compact and well proven design
- MEMS sensing technology
- Approved for the EU, US and Canadian markets.

Benefits

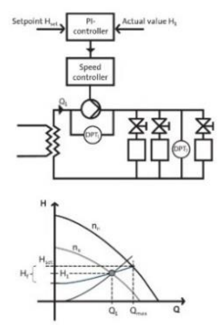
- Compatible with wet, aggressive media
- Accurate, linearised output signal
- Cost-effective and robust design.

Specifications

Pressure	
Measuring range (differential)	2.5 bar
Accuracy (IEC 61298-2)	2 % FS
Response time	< 0.5 s
Static Pressure P_1	16 bar
Static Pressure P_2	10 bar
Max system pressure	16 bar
Media and environment	
Media	Liquids, gasses and air
Media temperature (operation)	–10 to +70 °C
Media temperature (peak)	up to +80 °C
Ambient air temperature	–40 to +70 °C
Ambient air temperature (peak)	–55 to +90 °C
Humidity	0 to 95 % (relative), non-condensing
System burst pressure	25 bar
Electrical data	
Power supply	12-30 VDC
Output signals	4-20 mA
Load impedance	24 V max. 500 kΩ 16 V max. 200 kΩ 12 V max. 100 kΩ
Sensor materials	
Sensing element	Silicon-based MEMS sensor
Seal	FKM rubber
Housing	DIN W.-Nr. 1.4305
Wetted materials	FKM and PPS
Environmental standards	
Enclosure class	IP55
Temperature cycling	IEC 68-2-14
Vibration (non-destructive)	20 to 2000 Hz, 10G, 4h
Immunity	EN 61000-6-2
Emission	EN 61000-6-3
Weight	550 g

Flow compensated differential pressure control

(SPR Reglung)



TM03 0411 5004

Fig. 2 SPR Reglung

If the equipment is used in a manner not specified by the manufacturer, the protection provided by the equipment may be impaired.

BE > THINK > INNOVATE >

GRUNDFOS

Dimensions [mm]

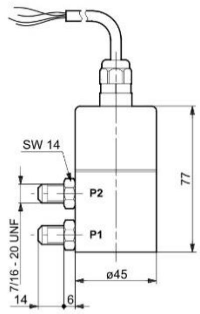


Fig. 3 Dimensional sketch

Output signals

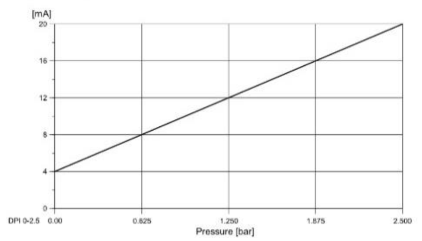


Fig. 4 Differential pressure response

Electrical connections

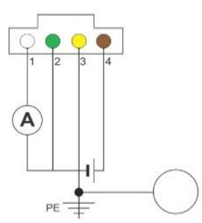


Fig. 5 Electrical connections

Pin configuration		Colour
1	Test conductor (can be cut off during mounting). Do not connect this conductor to the voltage supply.	White
2	Signal conductor	Green
3	GND (earth conductor)	Yellow
4	12-30 V supply voltage	Brown

Sensor Interface type SI 001 PSU

Power supply and amplifier for cables above 30 m and 2 wire connection of 400 VAC

Fig. 6 Sensor Interface, SI 001 PSU

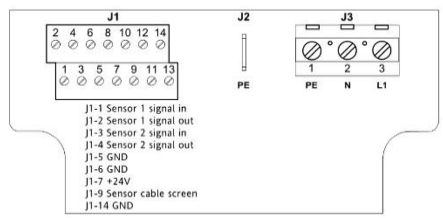


Fig. 7 Connections for power supply / amplifier

Part
Sensor Interface, SI 001 PSU

Accessories

Pos.	Component		
A	Fitting 6 mm	AISI 316	Tube connection
	Fitting 8 mm		
	Fitting 6 mm		Cutting ring
	Fitting 8 mm		
B	Cable for DPI 5.0 m		
	Cable for DPI 10.0 m		
	Wall bracket for sensor		

Type key

The DPI sensor is labelled with a type designation.

	96573683 - XX - XXX XXXXX
Product number	96573683
Version	- XX
Production year and week	- XXX
Consecutive number	XXXXX

For more information, see http://www.grundfos.com/directsensors.

96985463 1109 GB

Subject to alterations.

Grundfos Sensor A/S
Poul Due Jensens Vej 7. DK-8850 Bjerringbro. Denmark
Telephone: +45 87 50 14 00

www.grundfos.com\directsensors

B6. Siemens Flow Meter and Transmitter

The SITRANS F M MAG 5100 W with its patented liners of hard rubber NBR or ebonite and EPDM is a sensor for all water applications such as ground water, drinking water, cooling water, waste water, sewage or sludge applications. Application examples: Water abstraction, Water distribution network, Waste water and as custody transfer water meter or cooling meter.

Details

Measuring range	0 to 10 m/s
Nominal Sizes	From DN 15 to DN 2000 (1" to 78")
Accuracy	0.2 % ±2.5 mm/s
Operating Pressure	Max. 16 bar (Max. 150 psi)
Ambient temperature	From -40 to 70 °C (-40 to 158 °F)
Medium Temperature	From -10 to 70 °C (14 to 158 °F)
Liners	EPDM NBR hard rubber Ebonite hard rubber
Electrodes	Hastelloy C-276 Built-in grounding electrodes
Material	Carbon steel, with corrosion resistant two-component epoxy coating
Drinking Water Approvals	EPDM: WRAS, NSF/ANSI Standard 61, DVGW 270, ACS and BelgAqua NBR: NSF/ANSI Standard 61, WRAS Ebonite: WRAS
Custody Transfer Approvals	OILM R 49 MI-001 PTB K7.2 (Germany) BEV OE12/C040 (Austria)
General approval	MCERTS Sira Certificate No. MC080136/00

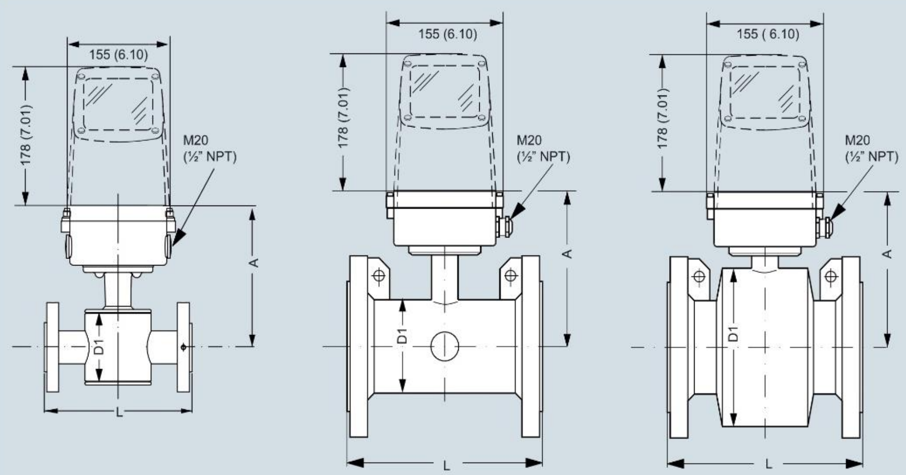
The SITRANS F M MAG 5000 is a microprocessor-based transmitter engineered for high performance, easy installation, commissioning and maintenance. The transmitter is truly robust, cost-effective and suitable for all-round applications and has a measuring accuracy of ± 0.4% of the flow rate (incl. sensor).

Application Examples: Water and waste water, General process industry, Food & beverage industry

Details

Accuracy	0.4 % ±1 mm/s
Input / output	1 current output 1 digital output 1 relay output
Communication	HART
Display	Background illumination with alphanumeric text, 3 x 20 characters
Enclosure	IP67 (NEMA 4x/6) IP20 (NEMA 2)
Power supply	12-24 V a.c./d.c. 115-230 V a.c.
Ambient temperature	From -20 to 50 °C (-4 to 122 °F)
Approvals	MI-001 Danak PTB OIML R49
Ex-approvals	FM/CSA Class 1, Div 2

Dimensional drawings



7ME6520: DN 15 ... 40 (½" ... 1½")

7ME6520: DN 50 ... 300 (2" ... 12")

7ME6520: DN 350 ... 1200 (14" ... 48")
7ME6580: DN 25 ... 2000 (1" ... 78")

		7ME6520 NBR or EPDM liner				7ME6580 Ebonite liner					
Nominal size		A		D1		A		D1		L	
[mm]	[inch]	[mm]	[inch]	[mm]	[inch]	[mm]	[inch]	[mm]	[inch]	[mm]	[inch]
15	½	177	7.0	77	3.0	-	-	-	-	200	7.9
25	1	187	7.4	96	3.8	187	7.4	104	4.09	200	7.9
40	1½	202	8.0	127	5.0	197	7.8	124	4.88	200	7.9
50	2	188	7.4	76	3.0	205	8.1	139	5.47	200	7.9
65	2½	194	7.6	89	3.5	212	8.3	154	6.06	200	7.9
80	3	200	7.9	102	4.0	222	8.7	174	6.85	200	7.9
100	4	207	8.1	114	4.5	242	9.5	214	8.43	250	9.8
125	5	217	8.5	140	5.5	255	10.0	239	9.41	250	9.8
150	6	232	9.1	168	6.6	276	10.9	282	11.1	300	11.8
200	8	257	10.1	219	8.6	304	12.0	338	13.31	350	13.8
250	10	284	11.2	273	10.8	332	13.1	393	15.47	450	17.7
300	12	310	12.2	324	12.8	357	14.1	444	17.48	500	19.7
350	14	382	15.0	451	17.8	362	14.3	451	17.76	550	21.7
400	16	407	16.0	502	19.8	387	15.2	502	19.76	600	23.6
450	18	438	17.2	563	22.2	418	16.5	563	22.16	600	23.6
500	20	463	18.2	614	24.2	443	17.4	614	24.17	600	23.6
600	24	514	20.2	715	28.2	494	19.4	715	28.15	600	23.6
700	28	564	22.2	816	32.1	544	21.4	816	32.13	700	27.6
750	30	591	23.3	869	34.2	571	22.5	869	34.21	750	29.5
800	32	616	24.3	927	36.5	606	23.9	927	36.5	800	31.5
900	36	663	26.1	1032	40.6	653	25.7	1032	40.63	900	35.4
1000	40	714	28.1	1136	44.7	704	27.7	1136	44.72	1000	39.4
	42	714	28.1	1136	44.7	704	27.7	1136	44.72	1000	39.4
	44	765	30.1	1238	48.7	755	29.7	1238	48.74	1100	43.3
1200	48	820	32.3	1348	53.1	810	31.9	1348	53.07	1200	47.2
1400	54	-	-	-	-	925	36.4	1574	65.94	1400	55.1
1500	60	-	-	-	-	972	38.2	1672	65.83	1500	59.1
1600	66	-	-	-	-	1025	40.4	1774	75.39	1600	63
1800	72	-	-	-	-	1123	44.2	1974	77.72	1800	70.9
2000	78	-	-	-	-	1223	48.1	2174	85.59	2000	78.7

- not available

B7. Cadillac Magnetic Flow Meter

Cadillac Meter

ACCURATE & RELIABLE ENERGY METERS

GENERAL INFORMATION

Cadillac® Magnetic Flow Meter
CMAG Series

CENTRAL STATION STEAM CO.® 15615 SW 74TH AVE., STE #150 TIGARD, OR 97224 PHONE: 888-556-3913 FAX: 503-624-6131 @ WWW.CADILLACMETER.COM

Rev 0613

METER INSTALLATION

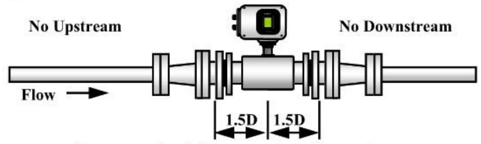
Cadillac CMAG Piping Requirements

Installation requirements have been redefined with the Cadillac magnetic flow meter.

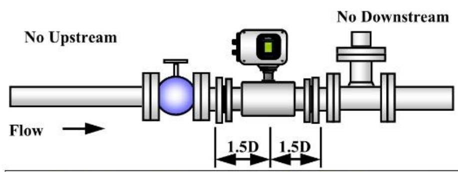
Employing coil and plate shaping techniques the Cadillac® meter provides a uniform magnetic flux shaping within the flow tube. This allows the meter to measure and sample uniformly the entire flow tube area.

In addition, the electronics provide high frequency DC square wave signal generation and flow signal sampling. When combined these two techniques eliminate all straight run flow profiling requirements, significantly decrease signal to noise ratio, and increase low flow accuracy.

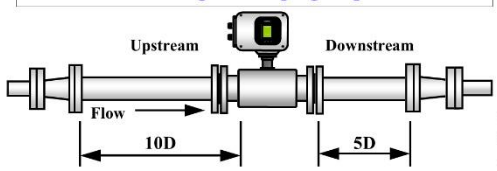
In practice, this means the Cadillac magnetic flow meters may be installed next to elbows, tees, valves, etc. without any effect in meter accuracy or stability. (See Illustrations). This also allows the CMAG to be installed in gravity flow applications with a turndown of 300:1 @ +/- 0.25% accuracy.



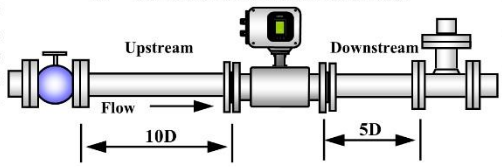
D = nominal flow meter diameter



Traditional Magmeter Piping Requirements



D = nominal flow meter diameter



In comparison, the straight pipe run requirements for all other magnetic flow meters are as follows:

Downstream of the meter:
- ◊ Expander (2-5) diameters
- ◊ Tee - (2-5) diameters
- ◊ Elbow - (2-5) diameters
- ◊ Valves - (2-10) diameters

• Upstream of the meter:
- ◊ Expander - (10) diameters
- ◊ Tee - (5) diameters
- ◊ Elbow - (5) diameters
- ◊ Valves - (10) diameters

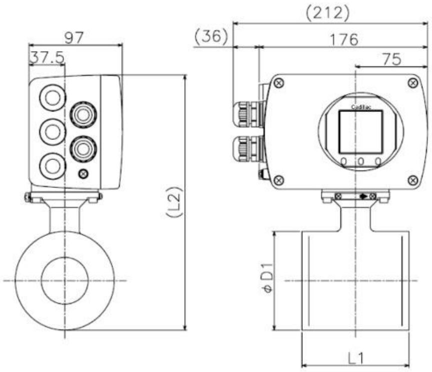
Unlike other technologies such as the Cadillac® Vortex flow meter, magnetic flow meters do not have a low flow cutoff, essentially allowing the meter to read to zero. With such a wide flow range capability for the technology, most applications can be addressed with meters at full line size.

The Cadillac® magnetic flow meter has a 304 stainless steel body and is always sold with integral grounding rings installed. The primary reason for providing grounding rings is to contain the magnetic field within the meter body and to assure the liquid potential is grounded properly. As a consequence, the induced voltage is remarkably free of noise allowing the meter to reliably measure extremely low fluid velocities. The table below lists minimum and maximum 4-20 madc output spans for each meter size, in GPM for liquids and lbs/hr for condensate.

CMAG	Liquid Flow Range Table		Condensate Flow Range Table	
Meter Body (inches) Size	Minimum Volumetric (gal/min) Range	Maximum Volumetric (gal/min) Range	Minimum Condensate (lbs/hr) Range	Maximum Condensate (lbs/hr) Range
0.50"	0.00 - 0.250	0.00 - 25.00	0.00 - 125.0	0.00 - 12,500
1.0"	0.00 - 0.750	0.00 - 75.00	0.00 - 375.0	0.00 - 37,500
1.5"	0.00 - 1.750	0.00 - 175.0	0.00 - 875.0	0.00 - 875,000
2.0"	0.00 - 3.000	0.00 - 300.0	0.00 - 1,500	0.00 - 150,000
3.0"	0.00 - 8.000	0.00 - 800.0	0.00 - 4,000	0.00 - 400,000
4.0"	0.00 - 12.50	0.00 - 1,250	0.00 - 6,250	0.00 - 625,000
6.0"	0.00 - 25.00	0.00 - 2,500	0.00 - 12,500	0.00 - 1,250,000
8.0"	0.00 - 50.00	0.00 - 5,000	0.00 - 25,000	0.00 - 2,500,000
10.0"	0.00 - 75.00	0.00 - 7,500	0.00 - 37,500	0.00 - 3,750,000

Low velocity "Turndown Accuracy" of the CMAG has allowed it to address applications, which were not possible for volumetric flow meters in the past. Below is the turndown accuracy for the CMAG:

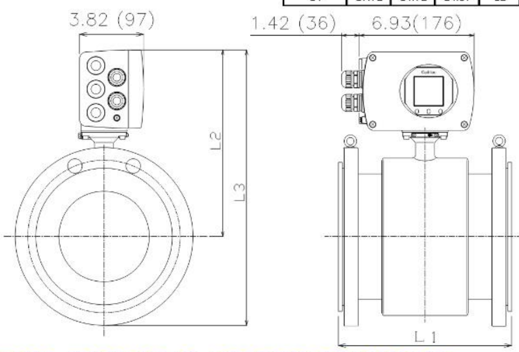
- ♦ **(+/- 0.25%)** of rate at **300:1** turndown with 1.5 diameters of straight piping from meter centerline up/downstream.
- ♦ **(+/- 0.50%)** of rate from **300:1 to 400:1** turndown with 1.5 diameters of straight piping from meter centerline up/downstream.
- ♦ **(+/- 1.00%)** of rate from **400:1 to 500:1** turndown with 1.5 diameters of straight piping from meter centerline up/downstream.



Cadillac Meter
ACCURATE & RELIABLE ENERGY METERS

Unit: inch (mm)

CMAG Flanged ANSI class 150 dimensions					
Meter Size (inch)	L1 (inch)	L2 (inch)	L3 (inch)	No. of Bolts	Weight (lbs)
1/2	5.51	7.99	9.88	4	21
1	6.30	8.23	10.71	4	25
1 1/2	6.69	8.58	11.34	4	28
2	7.09	9.13	12.20	4	32
3	9.06	10.04	13.66	4	62
4	9.45	10.31	14.45	8	72
6	10.24	11.22	16.73	8	115
8	11.81	12.20	18.70	8	145
10	13.78	13.18	21.06	12	210
12	15.75	14.37	23.15	12	355
14	17.72	14.72	24.37	12	460



CMAG Wafer ANSI class 150/300 dimensions				
Meter Size (inch)	L1 (inch)	L2 (inch)	D1 (inch)	Weight (lbs)
1/2	2.76	9.33	1.93	17
1	3.15	8.90	2.60	21
1 1/2	3.94	9.80	3.35	25
2	4.33	10.43	4.02	29
3	4.33	11.46	5.00	35
4	4.72	12.72	6.26	40

CADILLAC® MAGNETIC FLOW METER GENERAL SPECIFICATIONS

- Meter will consist of a full-bore body with encapsulated and rigidly retained set of coils.
- Meter available with remote or integral electronics with indication and totalization.
- Meter operates at ± 0.25% accuracy without flow profiling or piping straight run exceeding 1.5 diameters.
- Meter operates with minimum 300 to 1 turndown at stated operating accuracy.
- Meter available with pulse and analog (4-20 mA) outputs.
- Meter will provide instantaneous and totalized flow available at local indicator or remotely through outputs.
- Meter measures flow using Faraday's law (Induce voltage is directly proportional to the velocity of the conductive liquid)
- Meter K-factor is stable and not influenced by external piping or mounting orientation.
- Meter will have uniform magnetic field flux distribution eliminating piping straight run and flow profiling.
- Meter will measure fluids with conductivity greater than or equal to 3.0 uS/cm.
- Meter will be calibrated to/and provided with NIST calibration certificate.

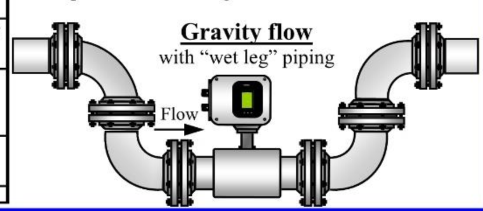
CADILLAC® MAGMETER MODEL NUMBER STRUCTURE

Code	Description
CMAG	Cadillac Magnetic Flow Meter
A	Size 0.5"
B	Size 1"
C	Size 1.5"
D	Size 2"
E	Size 3"
F	Size 4"
G	Size 6"
H	Size 8"
I	Size 10"
J	Size 12"
K	Size 14" (Consult factory for larger sizes)
II	Integral Converter with Indicator/Totalizer
RC	Remote Converter
W	Wafer Body (Ceramic Liner only) - available thru 4"
F	Flanged Body
150	ANSI Class 150
300	ANSI Class 300
S	Electrode Material - 316L SS (Std w/ Poly & EPDM)
H	Electrode Material - Hastelloy C (std w/PFA)
X	Electrode Material - Other
C	Liner Material - PFA (Teflon) - (Temp rated 250˚F)
U	Liner Material - Polyurethane (NSF approved)
D	Liner Material - EPDM (Rubber)
A	Liner Material - Ceramic (Temp rated 350˚F w/RC)
S	Ground Ring Material - 316 Stainless Steel
H	Ground Ring Material - Hastelloy C
X	Ground Ring Material - Other
FM	FM Approvals

Code	Description
CMC	CMAG Remote Converter
R	Remote Mounting
I	Indicator/Totalizer
U	Universal Mounting Bracket
XXFT	Interconnecting Cable (length in feet)
FM	FM Approvals

APPLICATION NOTE:

The Cadillac® magnetic flow meter is an excellent choice for gravity flow applications. However for proper flow meter operation the piping and flow tube must be full at all times. This requires the meter to be mounted in such a way to provide a "Wet Leg" as illustrated below.



B8. Omega FMG3002-PP Magnetic Flow Meter

SAFETY INSTRUCTIONS
1. Depressurize and vent system prior to installation or removal.
2. Confirm chemical compatibility before use.
3. Do not exceed maximum temperature/pressure specifications.
4. Wear safety goggles or face shield during installation/service.
5. Do not alter product construction.
6. Disconnect power before attempting any service or wiring.

2. Specifications

Wetted Materials:
- Sensor body, electrodes and grounding ring:
  - -PP: Polypropylene and 316L Stainless Steel
  - -PVDF and 316L Stainless Steel
- O-rings: FPM standard
  EPDM, (Perfluoroelastomer optional)

Other Materials:
- Case: PBT
- Ground terminal: 316 Stainless Steel

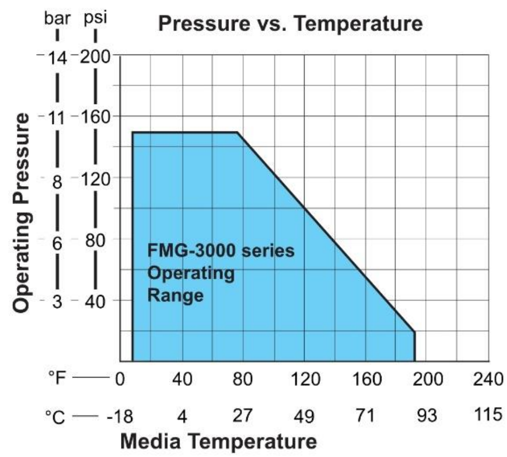
Power Requirements
- 4 to 20 mA: 21.6 to 26.4 VDC, 22.1 mA maximum
  400 mV p-p maximum ripple voltage
- Frequency: 5 to 26.4 VDC, 15 mA maximum
- Reverse polarity and short circuit protected

Performance
- Pipe Size Range: FMG-3000: ½ in. to 4 in.
  FMG-3100: 5 in. to 8 in.
  FMG-3200: 10 in. to 12 in.
- Flow Range: Minimum: 0.05 m/s (0.15 ft/s)
  Maximum: 10 m/s (33 ft/s)
- Linearity: ±(1% reading + 0.01 m/s)
  ±(1% reading + 0.033 ft/s)
- Repeatability ±0.5% of reading @ 25 °C
- Minimum Conductivity: 20 µS/cm

Output Specifications
Current output (4 to 20 mA)
- Programmable and eversible
- Loop Accuracy: 32 µA max. error
  (@ 25 °C @ 24 VDC)
- Temp. drift: ±1 µA per °C max.
- Power Supply Rejection: ±1 µA per V
- Isolation: Low voltage <48 VAC/DC
  from electrodes and auxiliary power
- Maximum Cable: 300 m (1000 ft)
- Maximum Loop Resistance: 300 Ω
- Error Condition: 22.1 mA

Frequency output:
- Max. Pull-up Voltage: 30 VDC
- Short Circuit Protected: ≤ 30 V @ 0 Ω pull-up for one hour
- Reverse Polarity Protected - 40 V
- Overvoltage Protected to 40 V with pull-up resistor
- Max. Current Sink: 50 mA, current limited
- Maximum cable: 300 m (1000 ft)

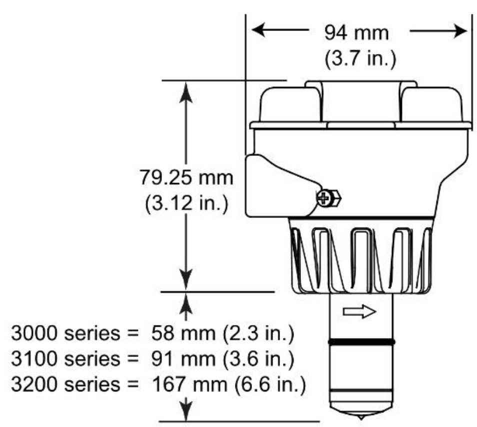
Environmental Requirements
- Storage Temperature: -20 to 70 °C (-4 to 158 °F)
- Relative Humidity: 0 to 95% (non-condensing)
- Operating Temperature
  Ambient: -10 to 70 °C (14 to 158 °F)
  Media: 0 to 85 °C (32 to 185 °F)



Max. Operating Pressure: 10.3 bar @ 25 °C (150 psi @ 77 °F)
1.4 bar @ 85 °C (20 psi @ 185 °F)

Tests, Approvals & Standards
- NEMA 4X
- CE

Dimensions



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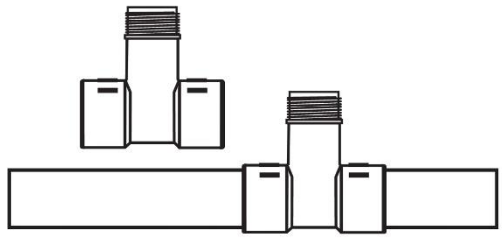
9. Calibration Data: K-Factors* and Full Scale Current Values

Plastic Installation Fittings: PVC Tees and Saddles



PIPE SIZE (IN.)	FITTING TYPE	K-Factor Gallons	K-Factor Liters*	20 mA= in GPM	20 mA= in LPM
SCH 80 PVC TEES FOR SCH 80 PVC PIPE					
½	FP-5305M	2277.00	601.58	13.10	49.60
¾	FP-5307M	1407.6	371.90	20.97	79.38
1	FP-5310M	861.17	227.52	34.21	129.50
1¼	FP-5312M	464.91	122.83	67.10	253.99
1½	FP-5315M	331.43	87.56	92.54	350.25
2	FP-5320M	192.89	50.96	145.15	549.38
SCH 80 PVC TEES FOR SCH 80 CPVC PIPE					
½	FP-5305CM	2277.0	601.58	13.18	49.87
¾	FP-5307CM	1407.6	371.90	21.31	80.67
1	FP-5310CM	861.17	227.52	34.84	131.86
1¼	FP-5312CM	464.91	122.83	64.53	244.24
1½	FP-5315CM	331.43	87.56	90.52	342.62
2	FP-5320CM	192.89	50.96	155.53	588.70
SCH 80 PVC SADDLES FOR SCH 80 PVC PIPE					
2	FP-5320S	193.83	51.21	154.77	585.81
2½	FP-5325S	138.01	36.46	217.38	822.78
3	FP-5330S	83.89	22.16	357.62	1353.60
4	FP-5340S	40.88	10.80	733.88	2777.74
6	FP-5360S	22.53	5.95	1331.85	5041.06
8	FP-5380S	12.52	3.31	2395.41	9066.64
10	FP-5381S	7.94	2.10	3778.75	14302.57
12	FP-5382S	5.71	1.51	5256.69	19896.57
SCH 80 PVC SADDLES FOR SCH 40 PVC PIPE					
2	FP-5320S	180.01	47.56	166.66	630.81
2½	FP-5325S	123.72	32.69	242.49	917.82
3	FP-5330S	75.81	20.03	395.71	1497.76
4	FP-5340S	41.87	11.06	716.56	2712.19
6	FP-5360S	19.71	5.21	1521.92	5760.46
8	FP-5380S	11.73	3.10	2558.12	9682.50
10	FP-5381S	7.43	1.96	4037.60	15282.3
12	FP-5382S	5.23	1.38	5734.87	21706.48

B9. Suresite Level Transducer and Visual Indicator

Standard Alloy Versions – Standard Size

ORDER IT!
Ordering is Easy! See Page D-9.
Easy online ordering too!

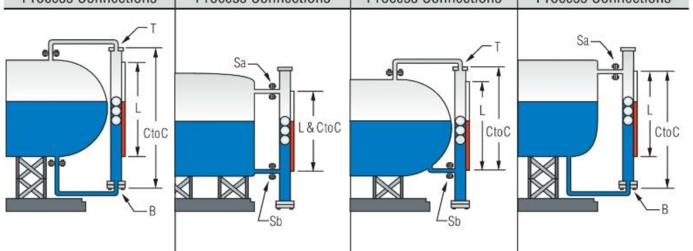
- Temperatures to 750°F (399°C)
- Pressures to 700 PSI (48 bar)

Rugged, welded construction makes these 2-1/2˝ (63.5 mm) diameter design, alloy SureSite Indicators dependable over a long service life indoors and out.

Type BA Shown

1. Mounting Configuration Types

To choose the best configuration for your application, focus on the process connections (connections where the liquid typically enters/leaves the SureSite).



	Type AA	Type BA	Type CA	Type DA
	Top and Bottom Process Connections	**Side and Side** Process Connections	**Top and Side** Process Connections	**Side and Bottom** Process Connections
Typical Lengths*	C to C = L + 10-1/4˝ (260.4 mm)	C to C = L	C to C = L + 3-3/4˝ (95.2 mm)	C to C = L + 6-1/2˝ (165.1 mm)
Flag Material	Plastic (300°F/148.9°C) or Aluminum (750°F/399°C)			
Length of Indication (Uninterrupted)	240˝ (610 cm)			
Minimum Specific Gravity	0.39			

* Dimensions vary due to connections, material and specific gravity.
Note: Additional materials, floats, connections and manufacturing techniques are available to extend lengths and operational capabilities. Please contact GEMS Sensors if the parameters above do not meet your requirements.

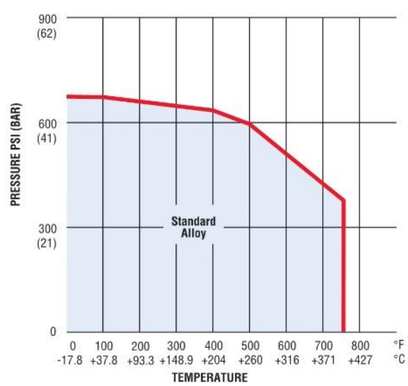
2. Material

Housing and Float: 316 Stainless Steel
Pressure/Temperature performance parameters for alloy SureSite versions are specified in the chart at right. Please consult the factory with temperature/pressure requirements that fall outside the parameters shown here.

= Stock Material (Best economy and delivery).

Materials		Code
Housing	**Float**	
316L Stainless Steel	316L Stainless Steel	2
Carpenter 20	Hastelloy C276	3*
Hastelloy C276	Hastelloy C276	4*

* Consult factory for pressure/temperature capabilities.



Note: SureSite Indicators are available for temperatures as low as -200°F (-129°C).

3. Connection Codes

(See complete descriptions below)

TOP — T

Blind		NPT				Flange	
		Fixed		Removable			
Fixed	Removable	Female	Male	Female	Male	Fixed	Removable
T1	T2	T3	T5	T6	T8	T9	T10

SIDE — Sa / SIDE — Sb

Blind	NPT		Flange
	Male	Female	
S1	S2	S3	S4

BOTTOM — B

Blind		NPT				Flange	
		Fixed		Removable			
Fixed	Removable	Female	Male	Female	Male	Fixed	Removable
B1	B2	B3	B5	B6	B8	B9	B10

—Connection Codes and Materials background-shaded in this color are stocked by Gems. Select these connections where possible to obtain the most economical SureSite Indicators with a prompt 3-day delivery.

Connection Code Descriptions

Please provide all connections when completing the OrderIt! Product Check List (located on the following page).
Note: Before selecting your connections, consider incorporating your vent and drain requirements.

T & B (Top and Bottom)

T/B 1. Welded pipe cap
T/B 2. Standard fixed flange/blind mating flange
T/B 3. Welded pipe cap w/FNPT
T/B 5. Welded pipe cap w/MNPT nipple
T/B 6. Standard fixed flange/mating FNPT reducing flange
T/B 8. Standard fixed flange/mating flange with MNPT nipple
T/B 9. Welded pipe cap with ANSI flange
T/B 10. Standard fixed flange/mating reducing flange spool

Sa & Sb Sides

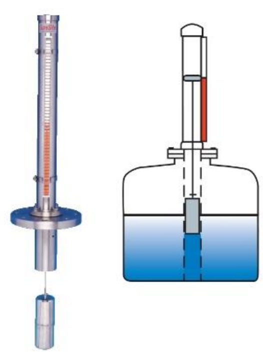
S1. No connection
S2. MNPT nipple
S3. FNPT coupling
S4. ANSI flange

3-Day
Qwik Configured
Available for up to 10 units
Built & Shipped in 3 Days!

Need it quick? Choose materials and components with the color shading for 3-Day manufacturing and shipping. See the Product Configurator section at www.gemssensors.com for further details.

Top Mount Units

When it's not practical to access the side of a tank for liquid monitoring, look to SureSite Top Mount Indicators for the solution. Please consult with the factory for these specially configured indicators **1-800-378-1600.**

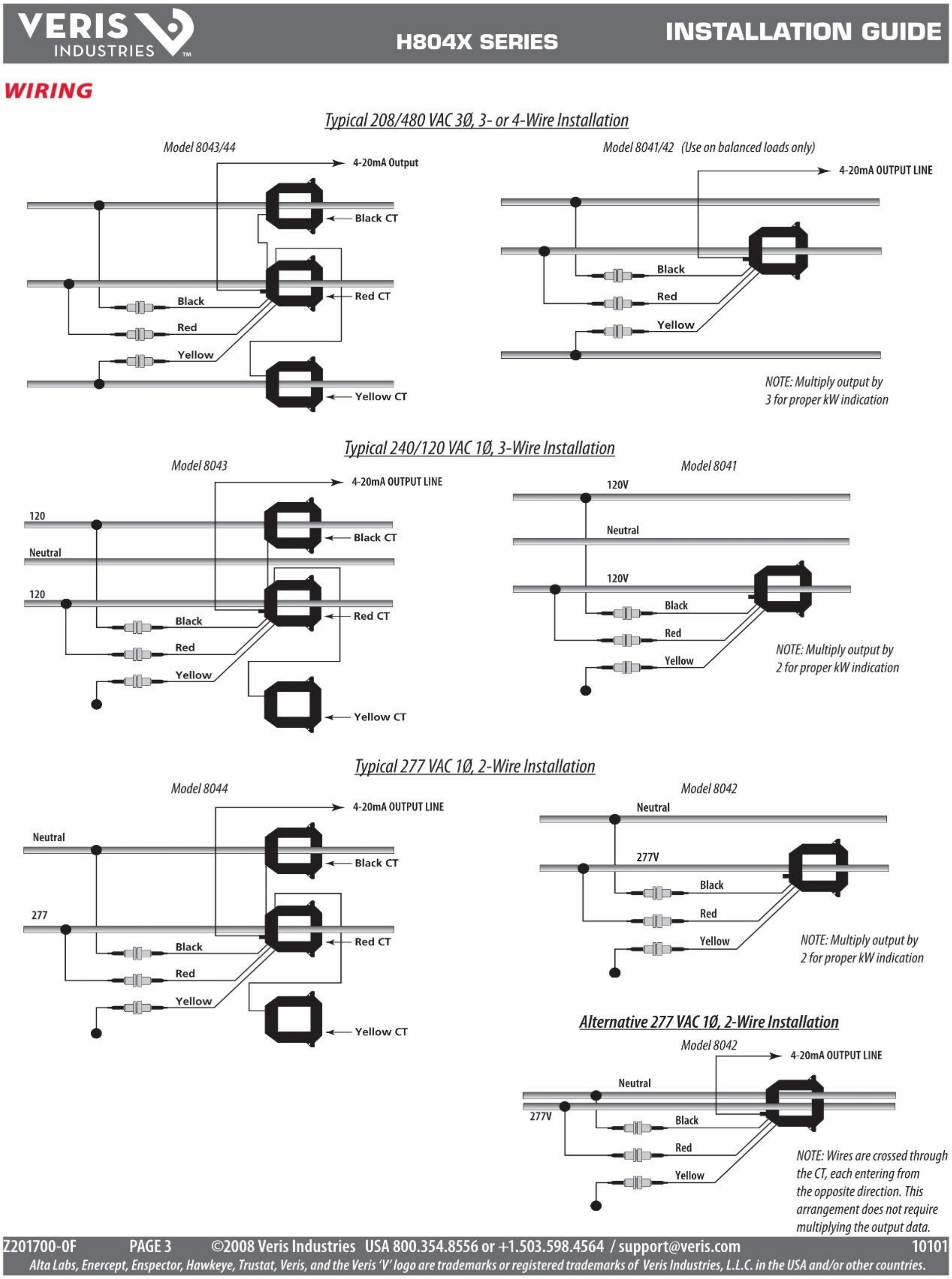


Accessories – Pages D-16 to D-18

Make more of your SureSite® Indicator with the productivity-enhancing accessories found at the end of this section.

- **Indicating Scales**
  Add graduations to your flag indication.
- **Switch Modules**
  Control pumps, valves, alarms, etc. Mount externally on housing for infinite positioning.
- **Continuous Output Transmitters**
  Signal conditioned for compatibility with most electronic instruments to 300°F (149°C).

B10. Power Monitor Sensor



TROUBLESHOOTING

Problem	Solution
Status LED does not blink	Check fuses and voltage connections. Status LED should blink regardless of CTs or output connections.
Readings seem highly inaccurate.	• Check that each CT is installed on the conductor with the corresponding color voltage input lead attached. In most cases, incorrect wiring will cause the STATUS LED to blink RED (slowly). However, a power factor lower than 0.5 could cause the LED to blink this way, even if the unit is installed properly. • It does not matter which side of the CT faces towards the load. • If current is below 7% of full scale maximum for the CT, use a smaller CT or wrap each wire through the CT multiple times • If using the single-phase H8042, use an amp-clamp to ensure that all three phases are passing the same approximate current. If phases are unbalanced, try the H8043/H8044 models.
Meter goes offline when load is switched off.	Voltage leads must be connected on the Line side of the conductor. The power meter cannot communicate without voltage.
Status LED blinks red.	• If the LED blinks quickly (i.e., about 5 blinks in two seconds), then use a higher rated CT. • If the LED blinks slowly (i.e., about 1 blink in two seconds) the CTs are not installed on the correct conductors, or the power factor is less than 0.5. The meter can accurately measure these low PFs, but few loads operate normally at such a low power factor.

NOTES

1. DO NOT GROUND THE SHIELD INSIDE THE ELECTRICAL PANEL. All wires, including the shield should be insulated to prevent accidental contact to high voltage conductors.
2. The cable should be mechanically secured where it enters the electrical panel.
3. The cable should be shielded twisted pair wire BELDEN 1120A or similar.

WARNING: After wiring the cable, remove all scraps of wire or foil shield from the electrical panel. This could be DANGEROUS if wire scraps come into contact with high voltage wires!

MAXIMUM READINGS

Model	3Ø Power (kW)	1Ø Power (kW)
H8041-0100-2	36.03	24.00
H8041-0300-2	108.1	72.00
H8041-0400-3	144.1	96.00
H8041-0800-3	288.2	192.0
H8041-0800-4	288.2	192.0
H8041-1600-4	576.4	384.0
H8041-2400-4	864.6	576.0
H8042-0100-2	83.14	55.43
H8042-0300-2	249.4	166.3
H8042-0400-3	332.6	221.7
H8042-0800-3	665.1	443.4
H8042-0800-4	665.1	443.4
H8042-1600-4	1330	886.7
H8042-2400-4	1995	1330
H8043-0100-2	36.03	36.03
H8043-0300-2	108.1	108.1
H8043-0400-3	144.1	144.1
H8043-0800-3	288.2	288.2
H8043-0800-4	288.2	288.2
H8043-1600-4	576.4	576.4
H8043-2400-4	864.6	864.6
H8044-0100-2	83.14	83.14
H8044-0300-2	249.4	249.4
H8044-0400-3	332.6	332.6
H8044-0800-3	665.1	665.1
H8044-0800-4	665.1	665.1
H8044-1600-4	1330	1330
H8044-2400-4	1995	1995

Appendix C: Calibration Data

C1. Cadillac Magnetic Flow Meter Calibration

The Cadillac Magnetic flow meter (CMAG series) was calibrated based on the Omega magnetic flow meter (FMG3002-PP) that has an accuracy of ±0.5% of reading.

Table C1.1: Cadillac flow meter calibration results

Flow Rate (GPM) (nominal)	**Average Error (%)**	**Standard Deviation (σ) (GPM)**
80	5.31	0.92
75	5.36	0.89
70	5.12	0.98
65	5.27	0.74
60	5.40	0.60
55	5.46	0.67
50	5.11	1.08
45	4.70	1.15
40	4.97	0.68
35	5.03	0.81
30	6.14	1.49
25	6.20	1.20
20	5.22	1.38
Overall Average Error	**5.33**	

$$Overall\ Average\ Error = \frac{\sum error\ of\ different\ flow\ rates}{Total\ number\ of\ errors} \quad (C1.1)$$

The Cadillac flow meter's reading was lower than that of the Omega flow meter by an overall average error 5.33%. The average errors of Cadillac flow meter shown in Table C1.1 were calculated for different flow rates in order to find the error in the Cadillac flow meter readings over a wide range of flow rates from 20 GPM to 85 GPM. For each flow rate, the data was gathered for 10-15 minutes, then the total error in the readings was calculated using Eq. (C1.2).

$$Error\ (\%) = 100 \sqrt{\frac{\sum_{i=1}^{Y} \left(\frac{Omega\ reading - Cadillac\ reading}{Omega\ reading}\right)^2}{Y}} \qquad (C1.2)$$

C2. Siemens Magnetic Flow Meter #1 Calibration

This magnetic flow meter (Model 3100) recoded the flow rate of the Worthington constant speed pump. This calibration was based on the Omega magnetic flow meter (FMG3002-PP) that has an accuracy of ±0.5% of reading.

Table C2.1: Siemens flow meter #1 calibration results, Test #1

Flow Rate (GPM) (nominal)	Average Error (%)	Standard Deviation (σ) (GPM)
85	2.31	0.79
80	2.60	0.73
75	3.03	0.92
70	3.13	1.04
65	2.12	0.94
60	2.19	1.65
55	1.85	1.11
50	2.17	1.82
45	2.51	1.64
40	2.06	1.21
35	1.81	1.32
Overall Average Error	**2.22**	

The average errors of Siemens flow meter #1, shown in Table C2.1, were calculated for different flow rates in order to find the error in Siemens flow meter #1 readings over a wide range of flow rates from 35 GPM to 85 GPM. For each flow rate, the data was gathered for 10-15 minutes, then the total error of readings was calculated using Eq. (C2).

$$Error\ (\%) = 100\sqrt{\frac{\sum_{i=1}^{Y}\left(\frac{Omega\ reading - Siemens\ reading}{Omega\ reading}\right)^2}{Y}} \qquad (C2)$$

Table C2.2: Siemens flow meter #1 calibration results, Test #2

Flow Rate (GPM) (nominal)	Average Error (%)	Standard Deviation (σ) (GPM)
85	2.54	0.90
80	2.91	0.80
75	2.78	0.86
70	2.62	0.68
65	2.36	0.70
60	1.77	0.77
55	1.61	0.90
50	2.29	1.56
45	2.47	1.65
40	1.77	1.24
35	1.31	1.11
Overall Average Error	**2.34%**	

The average errors of Siemens flow meter #1 shown in Table C2.1 was calculated for different flow rates in order to find the error in the Siemens flow meter #1 readings over a wide range of flow rates from 35 GPM to 85 GPM. For each flow rate, the data was gathered for 10-15 minutes, then the total error in the readings was calculated using Eq. (C2).

C3. Siemens Magnetic Flow Meter #2 Calibration

This magnetic flow meter (Model 3100) recorded the flow rate of the Grundfos variable speed pumps. This calibration was based on the Omega magnetic flow meter (FMG3002-PP) that has an accuracy of ±0.5% of reading. Even though this flow meter was not used to measure the flow rate of the Grundfos pumps (the Grundfos pumps' information was taken from the PC-Tools), the calibration was performed.

Table C3.1: Siemens flow meter #2 calibration results

Flow Rate (GPM) (nominal)	Average Error (%)	Standard Deviation (σ) (GPM)
85	1.36	0.85
80	1.77	0.88
75	2.00	0.91
70	1.07	0.79
65	1.19	1.11
60	1.26	1.09
55	2.09	1.46
50	2.12	1.93
45	1.45	1.23
40	2.24	0.86
35	2.01	0.96
Overall Average	**1.69**	

The average errors of Siemens flow meter #2, shown in Table C3.1, were calculated for different flow rates in order to find the error of the Siemens flow meter #2 readings over a wide range of flow rates from 35 GPM to 85 GPM. For each flow rate, the data was gathered for 10-15 minutes, then the total error of the readings was calculated using Eq. (C2).

Appendix D: Vent Condenser Energy Gain Calculations

Average gas Lower Heating value (BTU/ft^3) =1018.6 [55]

Water density at 160 ^oF (lb/ft^3) = 61

Water specific heat at 160 ^oF (Btu/(lb.F)) =1.0004

Temperature rise across the vent condenser (^oF) =19

Conversion factor of flow rate from GPM to ft^3/sec =0.00228

Example of calculating the natural gas savings from using the vent condenser in April of 2014

In order to calculate the natural gas savings for the vent condenser, the energy gain in the condensate has to be calculated.

Recall Eq. (10). In order to calculate the mass flow rate from pump discharge flow rate, Equation (D1) was used

$$m^o_{water} = \rho_{water}\, Q \tag{D1}$$

$$\mathrm{m}^o_{water} = 61\,\frac{lb}{ft^3}(\,89.8)\frac{gallon}{min}\ (0.00228)\ \frac{(\min)(ft^3)}{(s)\ (gallon)}\ (3600)\ \frac{sec}{hr}\ =\ 44961.7824\ {}^{lb}/_{hr}$$

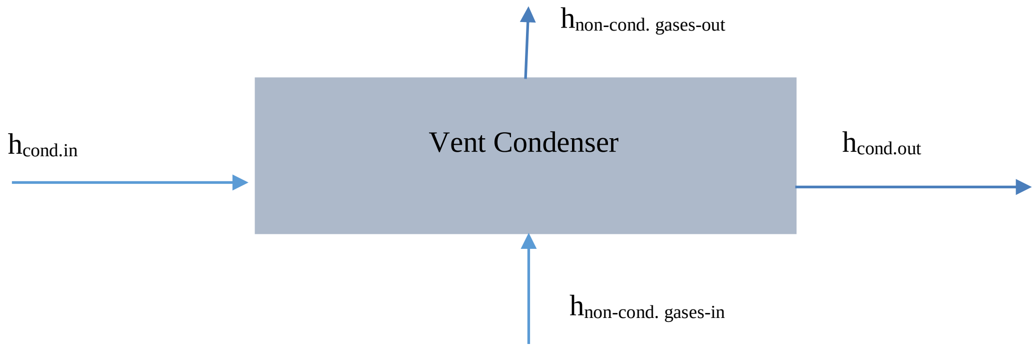


Figure D1: Simple diagram of vent condenser energy balance

From energy balance

$$h_{non-cond.gases-in} - h_{non-cond.gases-out} = h_{cond.out} - h_{cond.in} \qquad (D2)$$

Equation (D2) shows the energy balance between the non-condensable gases energy and the condensate water energy, where the condensate water gain energy represents the vent condenser energy savings in Eq. (10).

From Eq. (10)

$$E_{vent} = 854615.575 \frac{Btu}{hr}$$

Substituting into Eq. (12c):

$$\eta_{boiler} = \frac{E_{vent}}{m^{o}_{natural\ gas}\ (LHV\ fuel)}\ 100 \qquad (D3)$$

where the unknown is the amount of natural gas saving ($m^{o}_{natural\ gas}$)

$$m^{o}_{natural\ gas} = \frac{E_{vent}}{(\eta_{boiler})\ LHV\ fuel} = 927.494 \frac{ft^3}{hr} \qquad (D4)$$

The daily natural gas savings from the vent condenser can be calculated by multiplying the hourly natural gas savings by the total hours of a day (24 hr/day).

The natural gas savings for April of 2014 is obtained from

$$m^{o}_{natural\ gas}\ (from\ using\ the\ vent\ condenser\ in\ April, 2014) = 22259.857\frac{ft^3}{day}\ (30\frac{days}{month})$$

$$= 667{,}795.736\ \frac{ft^3}{month\ of\ April}$$

All other calculations were performed in an Excel sheet. These calculations are available in Ref. 62.

The boiler average efficiency can be found in Appendix F.

Table D1: Temperature rise across the vent condenser

Vent condenser flow rate (GPM)	ΔT (°F)	Date
91	17.5	01/26/2015
90	17.1	01/26/2015
N/A	19	03/25/2015
91.5	18	03/30/2015
95	20	04/01/2015
88.6	19	04/08/2015
Average	18.4	

Table D2: Annual vent condenser savings

		Vent Condenser Data available				
Month	Average month Boiler Efficiency	Data available from	Data available to	Average vent condenser flow rate (GPM)	Month's day	Natural Gas Saving (ft^3)
Apr-14	0.905	2-Apr-15	9-Apr-15	89.8	30	667,799.0
May-14	0.894	N/A	N/A	89.8	31	698,287.5
Jun-14	0.880	N/A	N/A	89.3	30	682,682.2
Jul-14	0.842	31-Jul-14	31-Jul-14	89.3	31	737,772.3
Aug-14	0.901	11-Aug-14	29-Aug-14	93.2	31	719,028.5
Sep-14	0.836	1-Sep-14	7-Sep-14	90.6	30	729,401.5
Oct-14	0.899	N/A	N/A	88.4	31	683,480.6
Nov-14	0.885	17-Nov-14	30-Nov-14	86.1	30	654,667.6
Dec-14	0.885	1-Dec-14	2-Dec-14	84.5	31	663,712.1
Jan-15	0.908	1-Jan-15	23-Jan-15	84.6	31	648,129.4
Feb-15	0.903	5-Feb-15	16-Feb-15	88.1	28	612,223.0
Mar-15	0.901	3-Mar-15	16-Mar-15	88.5	31	682,695.5
					Annual Natural gas saved in (ft^3)=	8,179,879.17
					Reclaimed Energy (BTU) =	8,332,024,923

Appendix E: Pressure Drop Calculations

E1. Friction Factor Estimate

Excel Sheet E1.1: Friction Factor Estimate

Pressure drop calculation from the pressure gauage at the discharge line to the DA tank

	water temp. at 160 F			
Dynamic Viscousity (Lbf-s/ft2)	0.00000838	Density slugs/ft3	61	4.253638555
Gravity (ft/s2)	32.2			
Kinematic Viscocity ft2/sec =	0.00000442			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f friction factor
3.826	0.318833333	0.07983941	5.89683714	425363.8555	0.4708	214	**0.02198742**

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor	Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257	1	1	8.583

hf (ft)	hbutv (ft)	helbs (ft)	hchecv (ft)	hTsec (ft)	Total
0.453050343	0.167081687	0.161984573	0.337845824	0.539948575	1.659911002

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)
3.826	0.318833333	0.07983941	2.94841857	212681.9278	0.2354	107

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor	Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257	1	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	hTsec (ft)	Total
0.628357404	0.034556709	0.121488429	0.084461456		0.784402542

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)
3.826	0.318833333	0.07983941	2.94841857	212681.9278	0.2354	107

From Tsection after the split on the **left** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	# T section no	K factor	Elevation z (ft)
67.5	2	0.3	2	0.256	0	0	3	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hTsec (ft)	Total
0.628357404	0.069113418	0.080992286	0.404961432	1.183424539

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q to DA tank (gpm)	Vent Condenser Flow Rate
3.826	0.318833333	0.07983941	3.634545882	262175.1988	0.29018	131.9	82.1

From Tsection after the split on the left line to the DA tank

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
22.08	3	0.3	0	0.256	1	0.625	10.8

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	Total
0.312337456	0	0.184610736	0.1282019	0.625150092

Change in Kinetic Energy (ft)
0.334825536

Δp-ΔZ +K.E	Eq. (14) = 0
4.252888175	0

delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
14.71806264	36.56	30.33	6.234734868

Excel Sheet E2.1: Pressure Drop Calculation, Trial Run #1

E2. Pressure Drop Calculations

	water temp. at 160 F			
Dynamic Viscousity (Lbf-s/ft2)	0.00000838	Density slugs/ft3	61	4.267552326
Gravity (ft/s2)	32.2			
Kinematic Viscocity ft2/sec =	0.00000442			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	5.91612586	426755.2326	0.47234	214.7	0.0218763082
						Pipe roughness (ft) e=	0.0004331782
						e/d =	0.0013586352

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257	1		1	8.583
hf (ft)	**hbutv (ft)**	**helbs (ft)**	**hchecv (ft)**	**hTsec (ft)**	**Total**					
0.453714609	0.168176533	0.163046018	0.340059645	0.543486727	1.668483531					

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.95806293	213377.6163	0.23617	107.35	0.0217946360
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257	1		1	8.583
hf (ft)	**hgatev (ft)**	**helb (ft)**	**hchecv (ft)**	**hTsec (ft)**	**Total**					
0.626929382	0.03478315	0.122284513	0.085014911		0.783997046					

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.95806293	213377.6163	0.23617	107.35	0.0217946360
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **left** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	# T section no	K factor	Elevation z (ft)
67.5	2	0.3	2	0.256	0	0	3	1	8.583
hf (ft)	**hgatev (ft)**	**helb (ft)**	**hTsec (ft)**	**Total**					
0.626929382	0.069566301	0.081523009	0.407615045	1.185633737					

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q to DA tank (gpm)	friction factor f Equa	Vent Condenser Flow Rate
3.826	0.318833333	0.07983941	3.652732389	263487.0686	0.291632	132.56	0.0221093781	82.14
						Pipe roughness (ft) e=	0.0004222820	
						e/d =	0.0013244599	

From Tsection after the split on the left line to the DA tank

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
22.08	3	0.3	0	0.256	1	0.625	10.8
hf (ft)	**hgatev (ft)**	**helb (ft)**	**hchecv (ft)**	**Total**			
0.317220846	0	0.186462865	0.129488101	0.633171812			

Change in Kinetic Energy (ft)
0.336305765

delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
14.73498036	36.56	30.32	6.241901403

Δp-ΔZ +K.E	Δp-ΔZ +K.E-toatal friction head=
4.271286127	0

Excel Sheet E2.2: Pressure Drop Calculation, Trial Run #2

	water temp. at 160 F			
Dynamic Viscousity (Lbf-s/ft2)	0.00000838	Density slugs/ft3	61	3.711771714
Gravity (ft/s2)	32.2			
Kinematic Viscocity ft2/sec =	0.00000442			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	5.145644843	371177.1714	0.41082525	186.73875	0.0219647066
						Pipe roughness (ft) e=	0.0004331782
						e/d =	0.0013586352

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257		1	1	8.583
hf (ft)	hbutv (ft)	helbs (ft)	hchecv (ft)	hTsec (ft)	Total					
0.344618782	0.127224338	0.123343141	0.257252677	0.411143802	1.26358274					

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.572822422	185588.5857	0.205412625	93.369375	0.0219603690
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257		1	1	8.583
hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	hTsec (ft)	Total					
0.477874045	0.026313203	0.092507355	0.064313169		0.596694604					

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.572822422	185588.5857	0.205412625	93.369375	0.0219603690
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **left** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	# T section no	K factor	Elevation z (ft)
67.5	2	0.3	2	0.256	0	0	3	1	8.583
hf (ft)	hgatev (ft)	helb (ft)	hTsec (ft)	Total					
0.477874045	0.052626407	0.06167157	0.308357852	0.900529873					

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q to DA tank (gpm)	friction factor f Equa	Vent Condenser Flow Rate
3.826	0.318833333	0.07983941	2.763468288	199340.6801	0.220633679	100.2880357	0.0223902916	86.45071429
						Pipe roughness (ft) e=	0.0004222820	
						e/d =	0.0013244599	

From Tsection after the split on the left line to the DA tank

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
22.08	3	0.3	0	0.256	1	0.625	10.8
hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	Total			
0.183873102	0	0.106724865	0.074114489	0.364712456			

delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
13.63295905	39.843125	34.07	5.775072932

Change in Kinetic Energy (ft)
0.292560619

Δp-ΔZ +K.E	Δp-ΔZ +K.E-toatal friction head
3.125519673	0

Excel Sheet E2.3: Pressure Drop Calculation, Trial Run #3

	water temp. at 160 F				
Dynamic Viscousity (Lbf-s/ft2)	0.00000838		Density slugs/ft3	61	3.326493106
Gravity (ft/s2)		32.2			
Kinematic Viscocity ft2/sec =		0.00000442			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	4.611531478	332649.3106	0.368181954	167.3554336	0.0220416598
						Pipe roughness (ft) e=	0.0004331782
						e/d =	0.0013586352

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257		1	1	8.583

hf (ft)	hbutv (ft)	helbs (ft)	hchecv (ft)	hTsec (ft)	Total
0.277759275	0.102183539	0.099066254	0.206619184	0.330220847	1.0158491

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.305765739	166324.6553	0.184090977	83.67771679	0.0221035227
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257		1	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	hTsec (ft)	Total
0.386318978	0.021134134	0.074299691	0.051654796		0.481752803

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.305765739	166324.6553	0.184090977	83.67771679	0.0221035227
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **left** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	# T section no	K factor	Elevation z (ft)
67.5	2	0.3	2	0.256	0	0	3	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hTsec (ft)	Total
0.386318978	0.042268268	0.049533127	0.247665636	0.725786009

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q to DA tank (gpm)	friction factor f Equa	Vent Condenser Flow Rate
3.826	0.318833333	0.07983941	2.15264178	155279.1752	0.17186565	78.12075017	0.0226994998	89.23468341
						Pipe roughness (ft) e=	0.0004222820	
						e/d =	0.0013244599	

From Tsection after the split on the left line to the DA tank

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
22.08	3	0.3	0	0.256	1	0.625	10.8

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	Total
0.113112161	0	0.064759006	0.044971532	0.222842699

delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
12.98796421	41.91018238	36.41	5.501845952

Change in Kinetic Energy (ft)
0.258266397

Δp-ΔZ +K.E	Δp-ΔZ +K.E-total friction head
2.446230611	0

Excel Sheet E2.4: Pressure Drop Calculation, Trial Run #4

	water temp. at 160 F				
Dynamic Viscousity (Lbf-s/ft2)	0.00000838		Density slugs/ft3	61	3.196390729
Gravity (ft/s2)		32.2			
Kinematic Viscocity ft2/sec =		0.00000442			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	4.431170002	319639.0729	0.353782	160.81	0.0220714412
						Pipe roughness (ft) e=	0.0004331782
						e/d =	0.0013586352

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor	Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257	1	1	8.583

hf (ft)	hbutv (ft)	helbs (ft)	hchecv (ft)	hTsec (ft)	Total
0.256803795	0.094346851	0.091468638	0.190773089	0.304895459	0.938287832

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.215585001	159819.5365	0.176891	80.405	0.0221586537
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor	Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257	1	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	hTsec (ft)	Total
0.357580963	0.019513309	0.068601478	0.047693272		0.445695751

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.215585001	159819.5365	0.176891	80.405	0.0221586537
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **left** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	# T section no	K factor	Elevation z (ft)
67.5	2	0.3	2	0.256	0	0	3	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hTsec (ft)	Total
0.357580963	0.039026619	0.045734319	0.228671595	0.671013495

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q to DA tank (gpm)	friction factor f Equa	Vent Condenser Flow Rate
3.826	0.318833333	0.07983941	1.951191766	140747.7318	0.155782	70.81	0.0228382718	90
						Pipe roughness (ft) e=	0.0004222820	
						e/d =	0.0013244599	

From Tsection after the split on the left line to the DA tank

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
22.08	3	0.3	0	0.256	1	0.625	10.8

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	Total
0.093500219	0	0.053205503	0.036948266	0.183653988

Change in Kinetic Energy (ft)	delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
0.245778234	12.79287283	42.62	37.20	5.419203075

Δp-ΔZ +K.E=	Δp-ΔZ +K.E-total friction head =
2.238651066	0

Excel Sheet E2.5: Pressure Drop Calculation, Trial Run #5

	water temp. at 160 F			
Dynamic Viscousity (Lbf-s/ft2)	0.00000838	Density slugs/ft3	61	2.140593911
Gravity (ft/s2)	32.2			
Kinematic Viscocity ft2/sec =	0.00000442			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.967514402	214059.3911	0.2369246	107.693	0.0224308081
						Pipe roughness (ft) e=	0.0004331782
						e/d =	0.0013586352

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257		1	1	8.583

hf (ft)	hbutv (ft)	helbs (ft)	hchecv (ft)	hTsec (ft)	Total
0.117048139	0.042313238	0.041022399	0.085559051	0.136741331	0.422684158

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	1.483757201	107029.6955	0.1184623	53.8465	0.0228129141
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257		1	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	hTsec (ft)	Total
0.165105154	0.008751445	0.0307668	0.021389763		0.204623399

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	1.483757201	107029.6955	0.1184623	53.8465	0.0228129141
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **left** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	# T section no	K factor	Elevation z (ft)
67.5	2	0.3	2	0.256	0	0	3	0.2	8.583

hf (ft)	hgatev (ft)	helb (ft)	hTsec (ft)	Total
0.165105154	0.01750289	0.0205112	0.0205112	0.223630444

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q to DA tank (gpm)	friction factor f Equa	Vent Condenser Flow Rate
3.826	0.318833333	0.07983941	2.967514402	214059.3911	0.2369246	107.693	0.0223125635	0
						Pipe roughness (ft) e=	0.0004222820	
						e/d =	0.0013244599	

From Tsection after the split on the left line to the DA tank

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
22.08	3	0.3	0	0.256	1	0.625	10.8

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	Total
0.211292763	0	0.123067198	0.085463332	0.419823293

Change in Kinetic Energy (ft)
0

delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
12.07076129	46.68175	41.57	5.113308603

Δp-ΔZ +K.E	Δp-ΔZ +K.E-total friction had
1.270761293	-5.9952E-15

Excel Sheet E2.6: Pressure Drop Calculation, Trial Run #6

	water temp. at 160 F				
Dynamic Viscousity (Lbf-s/ft2)	0.00000838	Density slugs/ft3	61	1.456175517	
Gravity (ft/s2)	32.2				
Kinematic Viscocity ft2/sec =	0.00000442				

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.018702284	145617.5517	0.161172	73.26	0.0229016745
						Pipe roughness (ft) e=	0.0004331782
						e/d =	0.0013586352

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257		1	1	8.583

hf (ft)	hbutv (ft)	helbs (ft)	hchecv (ft)	hTsec (ft)	Total
0.05530262	0.019581012	0.01898366	0.039593586	0.063278865	0.196739743

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	1.009351142	72808.77583	0.080586	36.63	0.0236434266
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor		Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257		1	1	8.583

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	hTsec (ft)	Total
0.079186138	0.004049847	0.014237745	0.009898396		0.09747373

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	1.009351142	72808.77583	0.080586	36.63	0.0236434266
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **left** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	# T section no	K factor	Elevation z (ft)
67.5	2	0.3	2	0.256	0	0	3	0.2	8.583

hf (ft)	hgatev (ft)	helb (ft)	hTsec (ft)	Total
0.079186138	0.008099695	0.00949183	0.00949183	0.106269492

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q to DA tank (gpm)	friction factor f Equa	Vent Condenser Flow Rate
3.826	0.318833333	0.07983941	2.018702284	145617.5517	0.161172	73.26	0.0227890603	0
						Pipe roughness (ft) e=	0.0004222820	
						e/d =	0.0013244599	

From Tsection after the split on the left line to the DA tank

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
22.08	3	0.3	0	0.256	1	0.625	10.8

hf (ft)	hgatev (ft)	helb (ft)	hchecv (ft)	Total
0.099866642	0	0.056950979	0.039549291	0.196366912

delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
11.39684988	48.39	43.56	4.827832239

Change in Kinetic Energy (ft)
0

Δp-ΔZ +K.E	Δp-ΔZ +K.E- total firction head
0.596849876	-6.43929E-15

E3. Pressure Drop to the Vent Condenser

Excel Sheet E3.1: Pressure Drop Calculation to the Vent Condenser

Dynamic Viscousity (Lbf-s/ft2)	0.00000838	Density slugs/ft3	61
Gravity (ft/s2)	32.2		
Kinematic Viscocity ft2/sec =	0.00000442		

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	Re	flow rate cfs	Total Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	5.89683714	425363.8555	0.4708	214	0.0218782620
						Pipe roughness (ft) e=	0.0004331782
						e/d =	0.0013586352

From discharge Pressure gauge to the Tsection before the split

Length (ft)	# Elbows No	k factor for Elbow	# ButterflyValve	K factor for Butterfly valve	# Check valve	K factor for check valve	T section no	K factor	Elevation z (ft)
12.167	1	0.3	1	0.30944	1	0.6257	1	1	8
hf (ft)	**hbutv (ft)**	**helbs (ft)**	**hchecv (ft)**	**hTsec (ft)**	**Total**				
0.450801141	0.167081687	0.161984573	0.337845824	0.539948575	1.6576618				

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.94841857	212681.9278	0.2354	107	0.0217983144
						Pipe roughness (ft) e=	0.0003758309
						e/d =	0.0011787693

From Tsection after the split on the **right** line to the Tsection where the 2 lines meet

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	T section no	K factor	Elevation z (ft)
67.5	3	0.3	1	0.256	1	0.6257	1	1	8
hf (ft)	**hgatev (ft)**	**helb (ft)**	**hchecv (ft)**	**hTsec (ft)**	**Total**				
0.622953133	0.034556709	0.121488429	0.084461456	0.134987144	0.778998271				

Change in Kinetic Energy (ft)		delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
0.494165678		9.942494393	36.56	32.3482489	4.211751097
Δp-ΔZ +K.E	Bernoulli Equation				
2.436660071	-5.77316E-15				

Pressure @ A 32.3482489 PSIG

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
3.826	0.318833333	0.07983941	2.262291258	163188.6567	0.18062	82.1	0.0220788491
						Pipe roughness (ft) e=	0.0003712834
						e/d =	0.0046503773

From Tsection after the split on the right line to the contrcation point to 2 in

Length (ft)	# Elbows No	k factor for Elbow	# Gate Valve	K factor for Gate valve	# Check valve	K factor for check valve	Elevation z (ft)
27.86	6	0.3	0	0.256	1	0.6257	16.66
hf (ft)	**hgatev (ft)**	**helb (ft)**		**Total**			
0.153322182	0	0.14304862		0.296370802			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
1.5	0.125	0.012271846	14.7182417	416239.8671	0.18062	82.1	0.0203380779
						Pipe roughness (ft) e=	0.0001209800
						e/d =	0.0098583356

From Tsection after the split on the right line to the contrcation point tobefore the heat Exchanger

Length (ft)	# Elbows No	k factor for Elbow	# Ball Valve	K factor for Gate valve	# Expansion	K factor for contraction	Elevation z (ft)
10.33266667	0	1.1	0	0.256	1	0.1914	39.75
hf (ft)	**hgatev (ft)**	**helb (ft)**	**hm contraction (2 inch)**	**Total**			
5.655074187	0	0	0.643825134	6.298899321			

Diameter (inch)	Diameter (ft)	Area	Velocity ft/s	RE	flow rate	Discharge Q (gpm)	friction factor f Equa
1.939	0.161583333	0.020506097	8.808112045	322000.9286	0.18062	82.1	0.0195545469
						Pipe roughness (ft) e=	0.0001209800
						e/d =	0.0058997077

From Tsection after the split on the right line to the contrcation point tobefore the heat Exchanger

Length (ft)	# Elbows No	k factor for Elbow	# Gate valve Valve Book	K factor for Gate valve Book	# Contraction	K factor for contraction	Elevation z (ft)
3.333333333	2	0.9	2	0.175	1	0.176	39.75
hf (ft)	**hgatev (ft)**	**helb (ft)**	**hm contraction (2 inch)**	**Total**			
0.485970307	0.421645858	2.168464411	0.212027631	3.288108206			

Change in Kinetic Energy (ft) =		delta P (ft)	Pressure P1 (psi)	Pressure P2 (psi)	Delta P from point 1 to 2
-0.664753875		52.73479228	36.56	14.22095605	22.33904395
Δp-ΔZ +K.E =	Δp-ΔZ +K.E - total friction haed=				
12.3200384	0				

Pressure before the Vent Condenser 14.22095605 PSIG

The calculations showed in Excel Sheet E3.1 (denoted by P (A)) that the pressure at point A in Fig. 45 is 32.34 PSIG.

The pressure before the vent condenser, i.e., before the water enters the vent condenser, is 14.22 PSIG (shown in Excel Sheet 3.1)

Kinematic Viscocity ft2/se	0.00000442
Gravity ft/s2	32.2
Density slugs/ft3	61
Kinematic Viscocity ft2/se	0.00000838

Input the calculated pressure Pressure at the inlet of (H.EX.) =	14.22095605

Pipe size	Diameter (ft)	Pipe Area(1.5 in) =	V(1.5 in)ft/s =	Re	Q (cfs)	Discharge Q GPM	friction factor f Equa
1.5	0.125	0.012271846	14.7182417	416239.8671	0.18062	82.1	0.0203380779
						Pipe roughness (ft) e=	0.0001209800
						e/d =	0.0000806533

From the vent condener to the 2" pipe							
Pipe Length (1.5in)	Kelbows (1.5 inch)		# 90 Deg. Elbow (1.5)	K elbow 45 (1.5 inch)	# 45 Deg. Elbow (1.5)	K expansion	Elevation
39	1.1		5	0.3	3	0.161243477	37.5
Hf (1.5in)	Helbow 90 deg (1.5 in)			Helbow 45 deg (1.5 in)		H expansion	
21.34472159	18.50072225			3.027390914		0.542385599	

Pipe ID inch	Diameter (ft)	Pipe Area (ft2)	V (2 in) ft/s =	Re	Q (cfs)	Discharge Q (gpm)	friction factor f Equa
						82.1	0.0195545469
1.939	0.161583333	0.020506097	8.808112045	322000.9286	0.18062	Pipe roughness (ft) e=	0.0001209800
						e/d =	0.0058997077

From the 2" pipe to the Storage tanks							
Pipe Length (2in)	Kelbows (2 inch)		#90 Deg. Elbow (2)	K elbow 45 (2 inch)	# 45 Deg. Elbow (2)	K valve	CV =
38.41666667	0.9		1	0.3	1	1.040359519	215
Hf (2 in)	Helbow 90 deg (2 in)			Helbow 45 deg (2 in)		Hvalve	
5.600807784	1.084232205			0.361410735		1.253323662	

Kienetic Energy (ft)	Total Hf (ft)
1.079532616	51.71499474

P1'(pressure before the storage tanks)=	6.435672294	psi	15.19240673 ft
P2 (Pressure at the outlet of the H.EX.)=	12	psi (Changible)	28.32786885 ft
Pressure drop across the H. EX.	2.22095605	PSi	
pressure in the condensate return	2	Psi (Assuming 10 ft pipe Height)	

The calculations showed that the pressure at Point B, P1', was 6.43 PSIG (shown in Excel sheet above).

Therefore, the pressure drop from points A to B was = 32.34 PSIG - 6.43 PSIG

= 25.91 PSIG

If the pressure required to move the water from point B to the storage tanks was 2 PSIG (see Fig. 45), the pressure at point A needed to increase another 2 PSIG so that condensate water could be driven from point A to the storage tanks (see Fig. 45). As a result, the overall pressure from point A to the inlet of the storage tank was roughly 25.91 + 2 = 27.91 PSIG ≈ 28 PSIG.

Appendix F: Monthly Steam Power Plant Steam Generation and Average Boiler Efficiencies

Table F1: April 2014 power plant data from Ref. 22

Makeup Water	%	apr 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
27,630	25.9	1	x		x				950,000	600,700		283,500		884,200		884,200	81	0.917761716
19,120	20.2	2	x						822,500	786,600				786,600		786,600	81	0.943020356
13,960	16.0	3	x		x	x			768,100	277,800		126,900	318,500	723,200		723,200	81	0.928418284
23,650	20.1	4			x	x			1,051,300			487,400	488,300	975,700		975,700	82	0.915150819
22,050	22.3	5			x	x			878,200			410,800	409,900	820,700		820,700	83	0.921497227
23,300	28.5	6			x	x			750,600			415,200	263,800	679,000		679,000	81	0.891998751
16,700	20.1	7			x				801,300			690,400		690,400		690,400	82	0.849588592
20,460	23.1	8	x		x				858,200	12,300		724,400		736,700		736,700	82	0.846457567
16,980	20.9	9			x				785,600			673,300		673,300		673,300	82	0.845104028
12,310	17.9	10			x				656,300			570,100		570,100		570,100	81	0.856547928
14,700	20.5	11			x				683,100			595,000		595,000		595,000	81	0.858886374
10,910	17.3	12			x				601,900			524,600		524,600		524,600	81	0.859423006
12,110	16.4	13			x	x			708,700			582,800	31,100	613,900		613,900	82	0.854158039
19,480	16.2	14			x	x			1,088,800			574,400	423,900	998,300		998,300	81	0.904099005
20,750	19.4	15			x				963,250			513,600	372,800	886,400		886,400	81	0.907389526
16,210	22.3	16			x				695,000			604,600		604,600		604,600	82	0.857800649
20,000	18.8	17	x		x	x			980,000	350,900		500,100	32,400	883,400		883,400	81	0.888862024
15,390	18.3	18	x		x				743,800	553,400		146,100		699,500		699,500	81	0.92733057
11,670	19.2	19	x						530,600	504,000				504,000		504,000	82	0.936626245
10,180	17.9	20	x						503,800	472,100				472,100		472,100	82	0.924014677
10,650	17.6	21	x		x				536,300	502,300				502,300		502,300	81	0.923545748
12,600	19.0	22	x						578,100	549,100				549,100		549,100	82	0.936594292
11,690	19.0	23	x						543,100	509,700				509,700		509,700	81	0.92541783
10,550	15.9	24	x						582,500	550,800				550,800		550,800	81	0.932397358
12,010	18.2	25	x						582,500	549,100				549,100		549,100	82	0.929519588
10,100	17.1	26	x						523,800	489,600				489,600		489,600	81	0.921677419
9,450	16.4	27	x						518,100	479,500				479,500		479,500	82	0.912594929
11,540	16.0	28	x						632,500	596,800				596,800		596,800	81	0.93040346
18,450	19.8	29	x			x			811,300	623,900			148,200	772,100		772,100	82	0.938415362
16,650	18.1	30	x			x			790,700	524,600			239,800	764,400		764,400	81	0.953261321
	#DIV/0!	31												0		0		
471,250	19.5								21,919,950	8,933,200	0	8,423,200	2,728,700	20,085,100	0	20,085,100		

Boiler Average Calculated Efficiency = 0.904598756

hsat.steam (Btu/lbm)	1197.7
hsat.water (Btu/lbm)	193.3
LHV Btu/CF	1018.6

Month average of steam produce (lbs)
669,503

Average steam generated when Grundfos CRE-3 and Worthington D-824 was in operation (lbs) in 2015
626,066.67

-6.48789401

The aveargе month steam generated was lower than the steam generated in the days in which the data was gathered in 6%
Therfore, monthly average energy consumption is eligible

Table F2: May 2014 power plant data from Ref. 22

In Table F2, it can be seen that there was no steam generated from May 19 to May 22. That was because the steam power plant was shut down for maintenance.

Makeup Water	%	may 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
15,160	18.4	1	x	x					798,800	503,600	178,900			682,500		682,500	82	0.842495581
13,030	17.7	2	x	x					745,100	101,500	510,100			611,600		611,600	81	0.80938648
11,420	19.1	3		x					611,300		497,400			497,400		497,400	81	0.802332561
9,190	16.5	4		x					576,300		462,900			462,900		462,900	81	0.79202993
9,470	17.1	5		x					571,900		458,500			458,500		458,500	81	0.790537135
9,680	16.4	6		x					608,800		488,700			488,700		488,700	82	0.791536101
9,130	15.7	7		x		x			551,900		201,700		280,000	481,700		481,700	82	0.860635556
9,630	15.6	8				x			528,100				511,000	511,000		511,000	82	0.954130469
12,920	19.1	9				x			578,100				562,600	562,600		562,600	81	0.959621105
9,830	16.1	10				x			517,500				505,300	505,300		505,300	81	0.962813068
10,290	18.1	11				x			485,600				472,500	472,500		472,500	82	0.959458439
11,110	16.1	12				x			592,500				573,600	573,600		573,600	82	0.954605254
12,630	16.9	13				x			638,800				619,100	619,100		619,100	82	0.955650142
13,130	18.4	14				x			613,800				591,500	591,500		591,500	81	0.950234725
15,510	20.1	15			x	x			671,900			23,600	615,600	639,200		639,200	81	0.938069806
16,410	19.6	16			x	x			728,200			97,100	596,800	693,900		693,900	82	0.939613494
12,720	17.1	17				x			635,000				618,200	618,200		618,200	81	0.959971429
6,900	14.6	18				x			396,900				391,600	391,600		391,600	81	0.972891965
0	######	19							0					0		0		
0	######	20							0					0		0		
0	######	21							0					0		0		
0	######	22							0					0		0		
18,510	45.1	23	x						388,100	340,400				340,400		340,400	81	0.864866232
19,280	32.9	24	x						528,100	486,500				486,500		486,500	82	0.908384488
16,240	30.5	25	x						491,900	442,300				442,300		442,300	81	0.886631484
17,540	31.0	26	x						503,100	469,400				469,400		469,400	82	0.920008416
17,900	33.0	27	x						488,800	450,200				450,200		450,200	81	0.908191276
18,090	34.8	28	x						476,900	431,800				431,800		431,800	81	0.892808565
15,880	29.3	29	x		x				520,000	175,400		274,300		449,700		449,700	82	0.852751665
18,770	33.6	30			x				536,900			464,200		464,200		464,200	81	0.852539999
17,490	32.8	31			x				510,600			442,300		442,300		442,300	82	0.854159865
367,860	22.1								15,294,900	3,401,100	2,798,200	1,301,500	6,337,800	13,838,600	0	13,838,600		
															Boiler Average Calculated Efficiency =			0.893939083

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F3: June 2014 power plant data from Ref. 22

Makeup Water	%	june 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
18,000	33.5	1			x				517,500			446,300		446,300		446,300	81	0.850392781
17,930	30.9	2			x				558,100			481,700		481,700		481,700	81	0.851074652
14,820	26.0	3			x				546,900			472,900		472,900		472,900	82	0.852637487
14,360	24.6	4			x				563,800			484,300		484,300		484,300	82	0.847017591
14,420	23.9	5			x				581,900			501,800		501,800		501,800	81	0.850325752
13,660	23.1	6			x				566,900			490,400		490,400		490,400	82	0.852996083
13,410	23.6	7			x				547,500			472,100		472,100		472,100	81	0.850262272
14,250	24.2	8			x				570,600			489,600		489,600		489,600	82	0.846082425
15,430	23.7	9			x				631,900			539,400		539,400		539,400	81	0.84171607
12,370	19.2	10			x				623,800			535,500		535,500		535,500	82	0.846480849
13,220	21.5	11			x				590,000			510,600		510,600		510,600	81	0.853359114
11,450	18.9	12			x	x			548,800			231,400	270,400	501,800		501,800	82	0.901611799
11,310	19.5	13	x			x			508,200	96,700			383,700	480,400		480,400	81	0.932119021
10,080	18.0	14				x			479,400				463,800	463,800		463,800	82	0.953972261
10,340	19.0	15				x			470,000				452,400	452,400		452,400	81	0.949134523
10,900	19.7	16				x			475,000				458,500	458,500		458,500	82	0.951806711
11,250	20.3	17				x			476,300				460,300	460,300		460,300	83	0.952935323
9,280	16.8	18				x			473,100				458,500	458,500		458,500	81	0.955629228
5,670	10.0	19				x			483,100				469,900	469,900		469,900	81	0.959116671
380	0.7	20				x			463,800				451,900	451,900		451,900	82	0.960759371
26,380	49.5	21				x			458,800				442,300	442,300		442,300	81	0.950597269
32,490	60.7	22				x			465,000				444,500	444,500		444,500	81	0.942587866
25,020	43.6	23				x			495,600				476,400	476,400		476,400	81	0.947858453
14,580	25.3	24				x			493,100				479,100	479,100		479,100	81	0.958063292
10,870	19.6	25				x			475,000				460,300	460,300		460,300	81	0.955543357
10,080	22.1	26		x		x			501,200		200,800		178,500	379,300		379,300	81	0.746233622
10,510	21.4	27		x					525,600		408,200			408,200		408,200	81	0.76580937
10,460	21.6	28		x					515,600		401,600			401,600		401,600	81	0.76803998
12,490	26.1	29		x					512,500		396,800			396,800		396,800	81	0.763450398
14,940	31.6	30		x					516,300		392,900			392,900		392,900	81	0.750382913
	######	31												0		0		
410,350	24.5								15,635,300	96,700	1,800,300	5,656,000	6,350,500	13,903,500	0	13,903,500		

Boiler Average Calculated Efficiency = 0.88026655

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F4: July 2014 power plant data from Ref. 22

Makeup Water	%	july 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
14,220	27.8	1		x					546,300		423,900			423,900		423,900	81	0.765130031
15,100	26.8	2		x					591,300		467,700			467,700		467,700	81	0.779942387
12,610	22.9	3		x					578,800		456,800			456,800		456,800	81	0.778216805
14,790	28.3	4		x					553,100		434,400			434,400		434,400	81	0.774442522
15,000	31.1	5		x					519,400		400,300			400,300		400,300	81	0.759952901
17,430	36.8	6		x					513,800		392,900			392,900		392,900	81	0.754034056
19,380	40.4	7		x					520,000		397,700			397,700		397,700	81	0.754145735
19,580	37.6	8		x					556,300		432,700			432,700		432,700	81	0.766974398
18,510	38.6	9		x					523,800		398,100			398,100		398,100	81	0.749427656
15,740	31.6	10	x	x					496,900	246,300	166,700			413,000		413,000	81	0.81956629
15,060	31.1	11	x						452,500	402,500				402,500		402,500	81	0.877102469
13,760	30.9	12	x						420,000	369,300				369,300		369,300	81	0.867027853
13,560	29.3	13	x						434,400	384,600				384,600		384,600	81	0.873016588
15,410	29.9	14	x						478,800	427,400				427,400		427,400	81	0.880204143
16,300	30.3	15	x						492,500	446,300				446,300		446,300	81	0.893559927
17,570	32.3	16	x						498,800	451,900				451,900		451,900	81	0.893344419
16,410	29.9	17	x						501,900	455,900				455,900		455,900	81	0.895685263
16,210	30.4	18	x						488,100	442,800				442,800		442,800	81	0.894544267
15,080	29.5	19	x						468,100	423,900				423,900		423,900	81	0.892951369
14,270	28.7	20	x						455,600	413,000				413,000		413,000	81	0.893859723
15,020	30.5	21	x						454,400	408,600				408,600		408,600	81	0.886672158
13,490	28.8	22	x						438,100	388,900				388,900		388,900	81	0.875321754
13,440	27.7	23	x						447,500	402,100				402,100		402,100	81	0.886021102
12,960	25.2	24	x		x				490,600	191,200		235,400		426,600		426,600	81	0.85742539
12,600	25.9	25	x		x				463,100	190,800		213,500		404,300		404,300	81	0.860858937
12,570	28.7	26	x						412,500	363,600				363,600		363,600	81	0.869166449
12,830	28.7	27	x						420,000	370,600				370,600		370,600	81	0.870079942
13,440	26.0	28	x		x				490,700	217,400		211,800		429,200		429,200	81	0.862475342
14,950	27.2	29			x				526,900			456,300		456,300		456,300	81	0.85393596
15,560	27.6	30			x				541,900			468,600		468,600		468,600	81	0.852680175
14,210	25.5	31			x				533,100			462,000		462,000		462,000	81	0.854547731
467,060	29.8								15,309,200	6,997,100	3,971,200	2,047,600	0	13,015,900	0	13,015,900		

Boiler Average Calculated Efficiency = 0.84168754

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F5: August 2014 power plant data from Ref. 22

Makeup Water	%	aug 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
14,170	27.7	1	x		x				482,500	168,900		255,500		424,400		424,400	82	0.867323452
12,680	27.8	2	x						425,000	378,900				378,900		378,900	82	0.879100865
12,880	27.1	3	x						438,100	394,600				394,600		394,600	82	0.888151104
12,650	24.8	4	x		x				486,900	167,600		255,500		423,100		423,100	82	0.856852924
13,210	25.0	5			x	x			505,600			402,500	35,900	438,400		438,400	82	0.855000783
10,520	19.7	6				x			464,400				444,100	444,100		444,100	81	0.942956361
13,380	24.2	7				x			481,900				459,400	459,400		459,400	81	0.940020006
13,510	24.4	8				x			480,000				460,300	460,300		460,300	82	0.94558978
12,940	23.9	9				x			469,400				448,900	448,900		448,900	82	0.942995352
13,790	25.2	10				x			472,500				453,300	453,300		453,300	81	0.945990856
14,850	26.0	11				x			496,300				473,800	473,800		473,800	82	0.941355823
16,520	28.1	12				x			511,300				487,400	487,400		487,400	81	0.939967341
15,060	25.8	13				x			509,400				484,300	484,300		484,300	82	0.937472551
14,450	25.6	14				x			493,100				469,000	469,000		469,000	81	0.937866174
13,530	24.9	15				x			475,600				451,900	451,900		451,900	82	0.936922196
11,920	22.7	16				x			458,100				435,300	435,300		435,300	81	0.936982345
12,380	23.1	17				x			469,400				445,400	445,400		445,400	82	0.935642972
13,670	25.0	18				x			478,100				454,100	454,100		454,100	81	0.936560399
14,420	27.1	19				x			465,600				442,300	442,300		442,300	81	0.936713976
12,400	24.0	20				x			450,600				428,300	428,300		428,300	81	0.937259647
10,780	21.1	21			x	x			480,000			237,600	185,900	423,500		423,500	81	0.869991901
11,120	21.7	22			x				489,400			424,800		424,800		424,800	81	0.855901082
10,970	22.6	23			x				454,400			402,100		402,100		402,100	81	0.872566997
10,320	21.2	24			x				455,600			404,300		404,300		404,300	81	0.875030232
11,210	22.0	25			x				481,300			422,600		422,600		422,600	81	0.865798169
12,900	24.4	26			x				503,800			438,400		438,400		438,400	81	0.858055569
12,430	23.0	27			x				520,000			449,300		449,300		449,300	81	0.851993158
11,250	21.1	28			x				512,500			442,800		442,800		442,800	82	0.851955233
12,060	22.0	29			x				522,500			455,000		455,000		455,000	81	0.858673646
10,530	19.8	30			x				496,300			441,400		441,400		441,400	82	0.87698282
11,400	21.7	31			x				495,600			435,800		435,800		435,800	82	0.867079584
393,900	24.0								14,925,200	1,110,000	0	5,467,600	7,059,600	13,637,200	0	13,637,200		
															Boiler Average Calculated Efficiency =			0.901443655

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F6: September 2014 power plant data from Ref. 22

Makeup Water	%	sep 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
11,780	20.4	1			x				540,000			479,900		479,900		479,900	81	0.876314549
13,100	23.8	2		x	x				567,500		282,600	174,600		457,200		457,200	89.1	0.794407596
12,120	23.0	3		x					569,400		436,600			436,600		436,600	82	0.7560827
12,610	26.0	4		x					585,000		402,900			402,900		402,900	82	0.679116736
13,420	25.4	5		x					564,400		439,300			439,300		439,300	81	0.767497961
12,200	22.8	6		x					563,800		443,200			443,200		443,200	82	0.775135652
13,140	24.5	7		x					569,400		444,900			444,900		444,900	81	0.770456237
13,010	23.8	8		x					580,600		454,100			454,100		454,100	82	0.771218613
12,590	24.9	9		x					548,800		419,100			419,100		419,100	82	0.753020137
13,360	24.0	10		x					594,400		462,000			462,000		462,000	81	0.766418902
13,280	21.1	11		x					646,900		521,500			521,500		521,500	82	0.794914088
15,980	22.8	12		x					723,100		581,900			581,900		581,900	81	0.793511139
15,460	22.8	13		x					709,400		563,500			563,500		563,500	81	0.783259676
13,140	22.6	14		x					613,800		481,700			481,700		481,700	81	0.773842886
14,480	23.3	15		x					644,400		514,900			514,900		514,900	80	0.787898715
11,250	18.3	16		x					649,400		511,400			511,400		511,400	82	0.7765179
9,050	15.6	17		x					614,400		480,400			480,400		480,400	81	0.771000792
9,540	16.0	18	x	x					571,900	308,400	185,100			493,500		493,500	82	0.850883482
9,500	16.0	19	x						523,800	492,200				492,200		492,200	81	0.926571947
9,380	17.5	20	x						481,300	444,500				444,500		444,500	82	0.910665609
9,160	17.0	21	x						483,100	448,400				448,400		448,400	81	0.915232848
9,740	16.6	22	x						516,900	486,900				486,900		486,900	82	0.928830087
10,920	18.5	23	x						524,400	489,600				489,600		489,600	81	0.920622868
9,440	15.8	24	x						525,000	497,400				497,400		497,400	82	0.934220751
9,090	16.1	25	x						505,000	469,900				469,900		469,900	81	0.917523294
8,800	15.7	26	x						503,800	466,400				466,400		466,400	81	0.912858389
8,240	15.3	27	x						483,100	445,800				445,800		445,800	81	0.909925967
7,810	14.6	28	x						476,900	443,200				443,200		443,200	82	0.916379703
8,750	15.9	29	x						491,900	458,100				458,100		458,100	82	0.918304054
8,510	15.9	30	x						480,000	445,400				445,400		445,400	81	0.914980856
	######	31												0		0		
338,850	19.8								16,851,800	5,896,200	7,625,100	654,500	0	14,175,800	0	14,175,800		

Boiler Average Calculated Efficiency = 0.835587138

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F7: October 2014 power plant data from Ref. 22

Makeup Water	%	oct 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
8,660	16.0	1	x						487,500	448,900				448,900		448,900	81	0.907983628
8,770	14.6	2	x			x			534,400	227,900			269,500	497,400		497,400	81	0.917787976
13,440	18.2	3				x			643,100				613,800	613,800		613,800	81	0.941133877
16,120	20.8	4				x			675,000				644,000	644,000		644,000	81	0.940773611
12,180	18.9	5				x			563,800				534,200	534,200		534,200	81	0.93429031
10,780	16.5	6				x			570,000				541,600	541,600		541,600	81	0.936929325
10,190	16.4	7				x			539,400				515,400	515,400		515,400	81	0.942185691
10,800	17.2	8				x			546,900				521,100	521,100		521,100	81	0.939541963
12,260	18.6	9				x			573,100				546,000	546,000		546,000	81	0.93943182
16,390	19.8	10				x			678,100				685,600	685,600		685,600	81	0.996965424
14,570	20.5	11				x			618,100				588,900	588,900		588,900	82	0.939476331
11,140	16.7	12				x			580,600				553,400	553,400		553,400	82	0.939864304
11,940	16.1	13				x			647,500				615,600	615,600		615,600	82	0.937479696
10,160	14.3	14				x			623,100				591,500	591,500		591,500	82	0.936052117
12,060	16.8	15				x			627,500				597,600	597,600		597,600	82	0.939074161
10,260	15.4	16			x	x			622,500			299,700	253,300	553,000		553,000	83	0.875969143
10,130	15.3	17			x				639,400			549,900		549,900		549,900	81	0.84803567
12,810	15.5	18			x				685,000			685,000		685,000		685,000	81	0.986059297
9,780	14.5	19			x				646,300			558,700		558,700		558,700	81	0.85240806
10,120	15.5	20			x				632,500			540,800		540,800		540,800	81	0.843100186
12,470	18.9	21			x				639,400			546,900		546,900		546,900	81	0.84340918
10,210	16.1	22			x				610,600			526,800		526,800		526,800	81	0.850730491
10,340	15.6	23			x				631,300			550,800		550,800		550,800	81	0.860322289
8,610	13.3	24			x				609,400			539,000		539,000		539,000	81	0.87214631
8,720	14.3	25			x				573,800			505,300		505,300		505,300	82	0.868343957
8,210	14.2	26			x				542,500			479,500		479,500		479,500	82	0.871549185
10,300	16.8	27			x				578,800			507,500		507,500		507,500	81	0.864590693
12,770	18.0	28			x				672,500			588,000		588,000		588,000	82	0.862160397
16,000	19.9	29			x				776,300			667,600		667,600		667,600	81	0.847988132
13,160	18.0	30		x	x				736,900		309,300	296,200		605,500		605,500	82	0.810230566
23,430	23.6	31		x					1,000,600		822,500			822,500		822,500	81	0.810547443
366,780	17.2								19,505,900	676,800	1,131,800	7,841,700	8,071,500	17,721,800	0	17,721,800		

Boiler Average Calculated Efficiency = 0.898598749

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F8: November 2014 power plant data from Ref. 22

Makeup Water	%	nov 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
20,110	20.2	1		x	x				995,100		772,600	52,900		825,500		825,500	81	0.81800015
15,430	18.5	2		x					838,600		690,700			690,700		690,700	82	0.812152583
16,470	19.2	3		x					868,800		710,900			710,900		710,900	81	0.806848014
18,460	18.5	4		x					997,500		827,800			827,800		827,800	82	0.81830565
13,380	15.1	5		x					884,400		736,800			736,800		736,800	82	0.821493091
19,310	18.6	6		x	x				1,036,300		839,100	21,400		860,500		860,500	82	0.81878223
16,410	17.5	7		x	x				945,500		683,800	96,000		779,800		779,800	81	0.813251232
17,540	17.9	8		x					977,500		814,600			814,600		814,600	81	0.821732894
15,160	17.1	9		x					892,500		733,700			733,700		733,700	81	0.810612556
19,900	22.4	10		x	x				894,500		717,000	20,700		737,700		737,700	82	0.813209551
41,000	28.6	11		x	x				1,411,300		678,100	511,000		1,189,100		1,189,100	82	0.830810678
35,880	22.1	12		x	x				1,585,700		705,700	642,700		1,348,400		1,348,400	82	0.838495526
29,200	17.4	13	x	x	x				1,563,100	429,200	322,000	643,100		1,394,300		1,394,300	82	0.879574229
32,750	19.8	14	x		x				1,494,400	713,100		658,000		1,371,100		1,371,100	82	0.904701487
25,160	16.4	15	x		x				1,391,900	645,800		630,900		1,276,700		1,276,700	82	0.904448527
30,410	18.4	16	x		x				1,503,200	697,800		674,600		1,372,400		1,372,400	82	0.900257969
32,500	17.6	17	x		x				1,688,200	779,200		752,900		1,532,100		1,532,100	82	0.894882981
31,130	17.5	18	x		x				1,633,800	752,100		724,900		1,477,000		1,477,000	81	0.891424643
29,670	20.0	19	x		x	x			1,305,700	632,600		257,300	341,300	1,231,200		1,231,200	81	0.929797202
29,110	18.9	20	x			x			1,400,800	651,900			625,100	1,277,000		1,277,000	81	0.89891328
24,090	17.1	21	x			x			1,213,100	624,800			546,000	1,170,800		1,170,800	81	0.951676057
14,250	15.4	22	x						795,000	765,600				765,600		765,600	81	0.949593708
15,370	15.8	23	x			x			856,200	696,900			112,000	808,900		808,900	81	0.931585337
26,150	19.2	24	x			x			1,144,400	651,900			476,900	1,128,800		1,128,800	81	0.972617734
25,070	17.3	25	x			x			1,243,100	667,200			532,400	1,199,600		1,199,600	81	0.951553964
28,840	20.5	26	x			x			1,206,900	654,100			515,200	1,169,300		1,169,300	81	0.955339412
31,980	20.6	27	x			x			1,323,200	694,800			596,800	1,291,600		1,291,600	81	0.962510723
21,540	18.5	28	x			x			1,005,000	698,700			268,200	966,900		966,900	81	0.948677348
14,500	17.1	29	x						738,800	705,300				705,300		705,300	81	0.941347621
36,080	26.7	30	x			x			1,161,900	725,400			395,500	1,120,900		1,120,900	81	0.951264193
	######	31												0		0		
726,850	19.1								34,996,400	12,186,400	9,232,800	5,686,400	4,409,400	31,515,000	0	31,515,000		

Boiler Average Calculated Efficiency = 0.884795352

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F9: December 2014 power plant data from Ref. 22

Makeup Water	%	dec 2014	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
34,730	20.3	1	x			x			1,456,300	728,900			692,600	1,421,500		1,421,500	81	0.962496251
31,480	20.1	2	x			x			1,330,000	669,800			629,100	1,298,900		1,298,900	82	0.96300182
25,720	17.7	3	x			x			1,221,900	625,200			578,400	1,203,600		1,203,600	81	0.971291407
25,400	18.6	4	x			x			1,163,700	590,200			544,700	1,134,900		1,134,900	82	0.961655664
21,350	17.3	5	x			x			1,047,500	537,300			487,800	1,025,100		1,025,100	82	0.96497316
20,950	16.1	6	x			x			1,095,600	565,300			513,600	1,078,900		1,078,900	82	0.971029003
20,850	16.2	7	x			x			1,091,300	561,300			508,800	1,070,100		1,070,100	81	0.966903742
22,780	19.4	8	x			x			994,400	525,000			448,900	973,900		973,900	81	0.965731244
28,900	19.9	9	x			x			1,230,000	637,000			568,300	1,205,300		1,205,300	81	0.966257944
22,510	15.1	10	x			x			1,260,700	650,100			584,500	1,234,600		1,234,600	81	0.965645124
18,700	13.4	11	x	x	x	x			1,266,400	260,300	359,200	39,800	496,600	1,155,900		1,155,900	81	0.900020484
18,290	15.8	12		x	x				1,136,900		703,900	257,700		961,600		961,600	81	0.834017609
12,700	13.6	13		x					926,900		775,300			775,300		775,300	81	0.824783443
14,020	17.3	14		x					806,900		671,600			671,600		671,600	82	0.820718086
29,100	24.5	15		x	x				1,181,300		717,100	270,400		987,500		987,500	82	0.824289813
27,680	18.1	16		x	x				1,483,800		665,000	603,300		1,268,300		1,268,300	81	0.842848771
30,610	19.1	17		x	x				1,538,800		696,100	631,800		1,327,900		1,327,900	82	0.85091509
29,100	18.8	18		x	x				1,486,900		678,600	602,900		1,281,500		1,281,500	81	0.849845308
29,050	19.4	19		x	x				1,445,000		661,900	580,100		1,242,000		1,242,000	82	0.84753332
25,310	19.3	20		x	x				1,267,500		598,100	489,600		1,087,700		1,087,700	81	0.846182799
22,450	17.9	21		x	x				1,214,400		578,400	462,000		1,040,400		1,040,400	81	0.844776097
21,450	18.3	22		x	x				1,150,000		548,200	423,900		972,100		972,100	81	0.833520211
24,470	18.6	23		x	x				1,282,500		598,500	493,100		1,091,600		1,091,600	82	0.839284467
24,230	17.6	24		x	x				1,336,900		620,400	521,900		1,142,300		1,142,300	82	0.842527889
22,590	18.6	25		x	x				1,188,100		564,800	445,800		1,010,600		1,010,600	81	0.838743814
21,460	18.4	26		x	x				1,143,200		548,200	422,200		970,400		970,400	81	0.837011846
25,640	18.4	27		x	x				1,351,200		626,500	530,300		1,156,800		1,156,800	82	0.844192862
31,520	21.7	28		x	x				1,406,900		646,200	558,700		1,204,900		1,204,900	81	0.844482797
30,620	20.1	29		x	x				1,386,300		642,700	621,900		1,264,600		1,264,600	81	0.899495482
31,740	17.6	30		x	x				1,760,600		768,200	729,100		1,497,300		1,497,300	82	0.838592858
28,280	14.1	31		x	x	x			1,899,400		841,800	450,600	367,900	1,660,300		1,660,300	81	0.861932321
773,680	18.1								39,551,300	6,350,400	13,510,700	9,135,100	6,421,200	35,417,400	0	35,417,400		

Boiler Average Calculated Efficiency = 0.884667765

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F10: January 2015 power plant data from Ref. 22

Makeup Water	%	jan 2015	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
22,210	13.2	1		x		x			1,537,500		743,300		651,400	1,394,700		1,394,700	81	0.894476034
23,140	15.4	2		x		x			1,370,000		678,100		567,400	1,245,500		1,245,500	81	0.896450259
23,600	17.4	3		x		x			1,252,500		623,400		504,900	1,128,300		1,128,300	81	0.888280004
28,080	15.1	4		x		x			1,713,800		796,700		744,600	1,541,300		1,541,300	82	0.886808959
28,690	15.2	5	x	x		x			1,745,100	441,000	798,400		329,900	1,569,300		1,569,300	82	0.88672446
29,030	15.5	6	x	x					1,705,000	765,400	787,500			1,552,900		1,552,900	81	0.898094711
48,200	22.0	7	x	x					2,003,700	928,800	888,100			1,816,900		1,816,900	82	0.894131425
35,560	17.1	8	x	x					1,921,900	835,200	894,700			1,729,900		1,729,900	81	0.88755085
30,390	14.8	9	x	x					1,873,700	814,600	885,900			1,700,500		1,700,500	82	0.894910516
28,670	14.7	10	x	x					1,780,600	765,600	851,800			1,617,400		1,617,400	82	0.895682527
28,440	16.4	11	x	x					1,572,500	659,800	777,400			1,437,200		1,437,200	82	0.901217438
33,190	17.8	12	x	x					1,702,500	725,400	822,900			1,548,300		1,548,300	81	0.896749257
31,310	15.8	13	x	x					1,815,700	782,700	862,300			1,645,000		1,645,000	82	0.893356581
30,610	18.4	14	x	x					1,528,700	630,400	752,900			1,383,300		1,383,300	81	0.892271751
27,540	19.7	15	x	x					1,281,900	546,400	614,300			1,160,700		1,160,700	81	0.892830194
29,560	25.0	16	x	x					1,058,800	700,400	282,200			982,600		982,600	82	0.915094319
28,430	25.5	17	x						954,400	925,300				925,300		925,300	81	0.955993994
26,840	23.6	18	x						978,800	945,000				945,000		945,000	81	0.952008618
29,030	26.2	19	x	x					976,900	892,500	27,100			919,600		919,600	81	0.928222059
25,800	26.8	20	x	x	x				948,200	687,600	111,200			798,800		798,800	81	0.830694122
28,020	27.7	21	x	x	x				1,113,800	672,900	167,600			840,500		840,500	82	0.744103824
37,410	27.5	22	x	x					1,255,000	681,600	447,600			1,129,200		1,129,200	81	0.887217656
25,950	19.7	23	x	x					1,222,500	564,400	530,700			1,095,100		1,095,100	82	0.883299416
20,000	18.4	24	x	x					969,400	646,200	255,100			901,300		901,300	81	0.916788987
25,780	22.0	25	x	x					1,060,000	782,700	190,300			973,000		973,000	81	0.905128015
26,270	22.7	26	x	x					903,900	764,300	195,600			959,900		959,900	81	1.047149374
30,560	28.5	27	x	x		x			948,200	760,400	55,600		75,300	891,300		891,300	82	0.92688742
17,240	19.6	28	x			x			781,200	270,800			459,700	730,500		730,500	81	0.922063897
39,650	32.1	29	x			x			1,042,500	426,100			598,900	1,025,000		1,025,000	82	0.969506743
35,800	25.9	30	x			x			1,156,300	582,300			563,500	1,145,800		1,145,800	82	0.9771052
29,600	22.6	31	x			x			1,092,500	553,000			534,600	1,087,600		1,087,600	81	0.981636697
904,600	19.9								41,267,500	18,750,800	14,040,700	0	5,030,200	37,821,700	0	37,821,700		
														Boiler Average Calculated Efficiency =				0.903176198

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

The calculated boiler efficiency on the 26th of January 2015 was more than 100%. That error might be due to steam power plant personal mistakenly typing the total natural gas consumption in that day. Therefore, this value was not used in calculating the average monthly boiler efficiency.

Table F11: February 2015 power plant data from Ref. 22

Makeup Water	%	feb 2015	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
35,500	22.6	1	x			x			1,318,800	657,100			647,500	1,304,600		1,304,600	81	0.975442037
45,950	23.5	2	x			x			1,660,000	845,300			778,300	1,623,600		1,623,600	81	0.964437274
35,470	23.1	3	x			x			1,302,500	695,600			577,100	1,272,700		1,272,700	81	0.963499169
47,000	25.5	4	x			x			1,539,400	789,700			742,900	1,532,600		1,532,600	82	0.981703572
44,070	23.4	5	x			x			1,589,400	808,500			754,300	1,562,800		1,562,800	82	0.969556732
37,160	27.1	6	x	x		x			1,192,000	389,800	111,600		635,700	1,137,100		1,137,100	81	0.940644318
24,570	24.0	7		x		x			973,700		566,600		284,800	851,400		851,400	81	0.862206928
24,110	25.5	8		x					930,000		786,200			786,200		786,200	81	0.833591204
45,910	35.2	9	x	x					1,230,700	399,000	682,500			1,081,500		1,081,500	82	0.866517535
45,050	36.2	10	x	x					1,167,500	499,600	532,400			1,032,000		1,032,000	81	0.871617297
42,235	30.8	11	x	x					1,263,800	560,900	578,800			1,139,700		1,139,700	82	0.8892323
43,175	26.1	12	x	x					1,503,100	680,300	690,400			1,370,700		1,370,700	81	0.899202634
35,360	24.7	13	x	x					1,313,800	569,200	621,300			1,190,500		1,190,500	82	0.89351773
34,280	24.2	14	x	x					1,298,200	559,600	615,100			1,174,700		1,174,700	81	0.892253779
45,040	23.2	15	x	x					1,775,700	811,600	801,900			1,613,500		1,613,500	82	0.895988442
43,250	22.8	16	x	x					1,742,600	790,100	785,800			1,575,900		1,575,900	81	0.891731233
38,940	22.1	17	x	x					1,607,500	724,900	736,300			1,461,200		1,461,200	82	0.896317166
44,630	22.8	18	x	x					1,785,000	815,100	809,800			1,624,900		1,624,900	81	0.897617788
43,520	21.6	19	x	x					1,841,900	834,300	835,600			1,669,900		1,669,900	82	0.893979272
41,440	24.0	20	x	x					1,587,500	697,400	738,100			1,435,500		1,435,500	82	0.89164606
30,920	23.5	21	x	x					1,210,600	500,900	592,800			1,093,700		1,093,700	82	0.890841775
48,620	26.7	22	x	x					1,647,500	741,100	770,400			1,511,500		1,511,500	81	0.904660775
47,720	24.7	23	x	x					1,759,400	796,700	808,900			1,605,600		1,605,600	81	0.899861775
38,600	24.3	24	x	x					1,471,400	630,000	686,900			1,316,900		1,316,900	81	0.88252106
37,800	26.9	25	x	x		x			1,312,500	531,100	631,800		3,100	1,166,000		1,166,000	81	0.875996297
48,970	25.0	26		x		x			1,810,100		816,800		812,000	1,628,800		1,628,800	81	0.887295389
47,490	22.5	27		x		x			1,941,900		898,600		855,800	1,754,400		1,754,400	81	0.89085042
46,910	23.4	28		x		x			1,841,300		868,900		792,300	1,661,200		1,661,200	81	0.889611527
	######	29												0		0		
	######	30												0		0		
	######	31												0		0		
########	24.9								41,617,800	15,327,800	15,967,500	0	6,883,800	38,179,100	0	38,179,100		

Boiler Average Calculated Efficiency = 0.90329791

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Table F12: March 2015 power plant data from Ref. 22

Makeup Water	%	mar 2015	#7	#8	#1	#2	HRS. on OIL	OIL Gals.	GAS FEET M.	No. 7 Steam Lbs.	No. 8 Steam Lbs.	No. 1 Steam Lbs.	No. 2 Steam Lbs.	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.	Caculated Boiler Efficiency
39,760	24.1	1		x		x			1,514,400		752,500		616,000	1,368,500		1,368,500	81	0.891060584
40,350	25.9	2		x		x			1,433,800		701,800		588,900	1,290,700		1,290,700	81	0.88764593
38,170	27.2	3		x		x			1,298,800		650,600		512,800	1,163,400		1,163,400	81	0.883262539
39,040	23.5	4		x		x			1,527,500		742,900		637,000	1,379,900		1,379,900	81	0.890777888
36,990	21.1	5		x		x			1,605,700		440,600	364,400	648,400	1,453,400		1,453,400	81	0.892531969
28,225	20.9	6			x	x			1,205,700			568,300	553,000	1,121,300		1,121,300	81	0.917034328
21,065	20.0	7			x	x			951,300			560,000	313,700	873,700		873,700	81	0.905623891
23,480	24.4	8			x	x			908,100			711,800	87,900	799,700		799,700	81	0.868353287
27,880	26.9	9			x	x			977,500			742,900	117,300	860,200		860,200	81	0.867732181
21,170	21.3	10			x	x			949,400			779,200	45,100	824,300		824,300	81	0.856128796
19,850	23.2	11			x	x			821,800			683,100	26,800	709,900		709,900	81	0.851793009
18,340	23.7	12			x				739,400			643,600		643,600		643,600	82	0.858301006
14,555	19.3	13			x				712,500			624,800		624,800		624,800	82	0.864687507
18,915	22.7	14			x				793,100			691,700		691,700		691,700	82	0.859988924
19,185	23.3	15			x				778,100			683,400		683,400		683,400	82	0.866049253
14,490	20.8	16		x	x				653,100			577,900		577,900		577,900	82	0.87252131
22,770	24.9	17		x					898,200		482,100	277,400		759,500		759,500	81	0.833792069
28,480	25.3	18		x	x				1,123,800		904,800	29,800		934,600		934,600	82	0.820048958
27,560	23.4	19		x	x	x			1,101,300		191,600	532,000	253,800	977,400		977,400	82	0.875124269
20,245	19.7	20			x	x			905,000			291,400	563,500	854,900		854,900	82	0.931471926
18,745	22.5	21				x			712,500				692,600	692,600		692,600	82	0.958518834
16,710	21.5	22				x			663,800				646,500	646,500		646,500	82	0.960360554
23,890	25.7	23	x		x	x			813,800	399,000		59,500	312,800	771,300		771,300	81	0.934563205
22,045	21.1	24	x						919,400	867,600				867,600		867,600	82	0.930503639
19,660	18.6	25	x						909,400	878,500				878,500		878,500	81	0.952554533
25,000	21.9	26	x						990,600	949,400				949,400		949,400	82	0.945048149
24,080	19.7	27	x						1,068,100	1,013,300				1,013,300		1,013,300	82	0.935468482
20,470	19.1	28	x						921,300	889,000				889,000		889,000	81	0.951488891
17,970	19.3	29	x						796,900	772,600				772,600		772,600	82	0.955991232
21,975	25.2	30	x						753,100	724,900				724,900		724,900	82	0.949136083
19,265	25.8	31	x						638,100	619,900				619,900		619,900	82	0.957934741
750,330	22.7								30,085,500	7,114,200	4,866,900	8,821,200	6,616,100	27,418,400	0	27,418,400		

Boiler Average Calculated Efficiency = 0.900822515

hsat.steam (Btu/lb)	1197.7
hsat.water (Btu/lb)	193.3
LHV Btu/CF	1018.6

Month average of steam produce (lbs)

884,465

Average steam generated when Grundfos CRE-3 and Worthington D-824 was in operation (lbs).

709,900

-19.7367

The avearge month generated steam was lower than the average generated steam in the day in which the data was gathered by 19.7%. Therefore, the daily average energy consumption is necessary.

Monthly average energy consumption is ineligible.

Appendix G: Grundfos Pumps Data Curve Fitting

A curve fit was made for the recorded data for the Grundfos pumps in order to calculate the theoretical power consumption from the pump curves and Eq. (18).

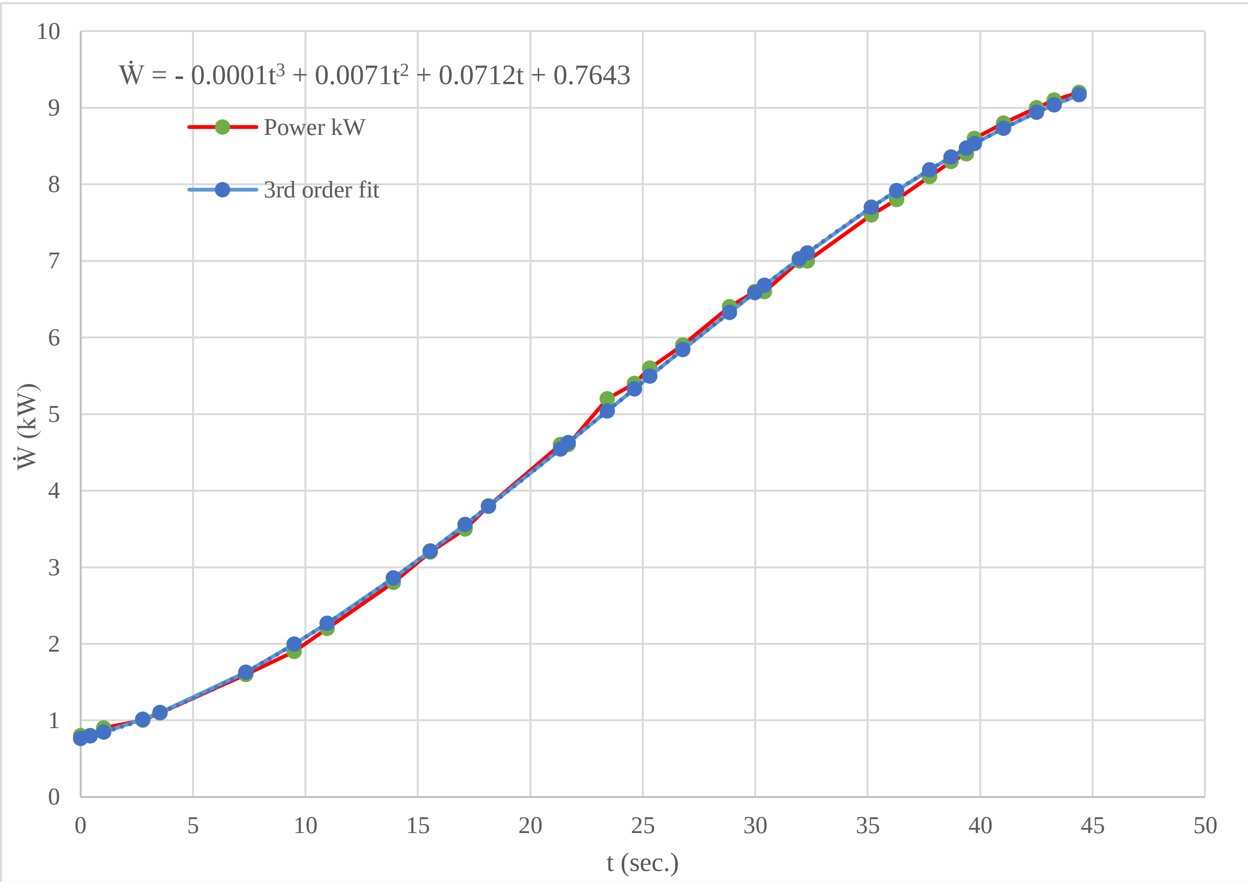


Figure G1: Increasing part of the first peak of power consumption of Grundfos pumps in Case 2, Test #2 (refer to Fig. 62 in main text for increasing peak)

Table G1: Grundfos pumps' power consumption comparison with the 3rd order equation of Fig. G1

Time Interval (sec)	Accumulated Time (sec)	Recorded Values (kW)	3rd Order Power Fit (kW)	Error (%)
0.00	0.000	0.800	0.764	4.463
0.432	0.432	0.800	0.796	0.453
0.604	1.036	0.900	0.846	6.047
1.729	2.765	1.000	1.013	-1.334
0.777	3.542	1.100	1.101	-0.102
3.802	7.344	1.600	1.631	-1.907
2.160	9.504	1.900	1.996	-5.077
1.469	10.973	2.200	2.268	-3.107
2.937	13.910	2.800	2.859	-2.118
1.642	15.552	3.200	3.213	-0.397
1.555	17.107	3.500	3.559	-1.700
1.037	18.144	3.800	3.796	0.100
3.197	21.341	4.600	4.545	1.186
0.345	21.686	4.600	4.627	-0.598
1.728	23.414	5.200	5.040	3.075
1.210	24.624	5.400	5.329	1.306
0.691	25.315	5.600	5.494	1.885
1.469	26.784	5.900	5.843	0.961
2.073	28.857	6.400	6.328	1.121
1.124	29.981	6.600	6.586	0.212
0.431	30.412	6.600	6.684	-1.266
1.556	31.968	7.000	7.029	-0.419
0.345	32.313	7.000	7.104	-1.492
2.852	35.165	7.600	7.699	-1.307
1.123	36.288	7.800	7.919	-1.525
1.468	37.756	8.100	8.192	-1.130
0.951	38.707	8.300	8.358	-0.705
0.691	39.398	8.400	8.475	-0.889
0.346	39.744	8.600	8.531	0.800
1.296	41.040	8.800	8.732	0.768
1.468	42.508	9.000	8.939	0.676
0.778	43.286	9.100	9.039	0.671
1.123	44.409	9.200	9.170	0.322
	7.344	N/A	1.631	
	17.107	N/A	3.559	
	35.165	N/A	7.699	

The increasing part of the first peak of power consumption lasted for 44.409 sec (see Fig. G1). Therefore, three points of time were selected in order to determine the Grundfos pumps' efficiency. These points in time were 7.344 sec, 17.107 sec, and 35.165 sec.

Flow Rate Curve Fitting

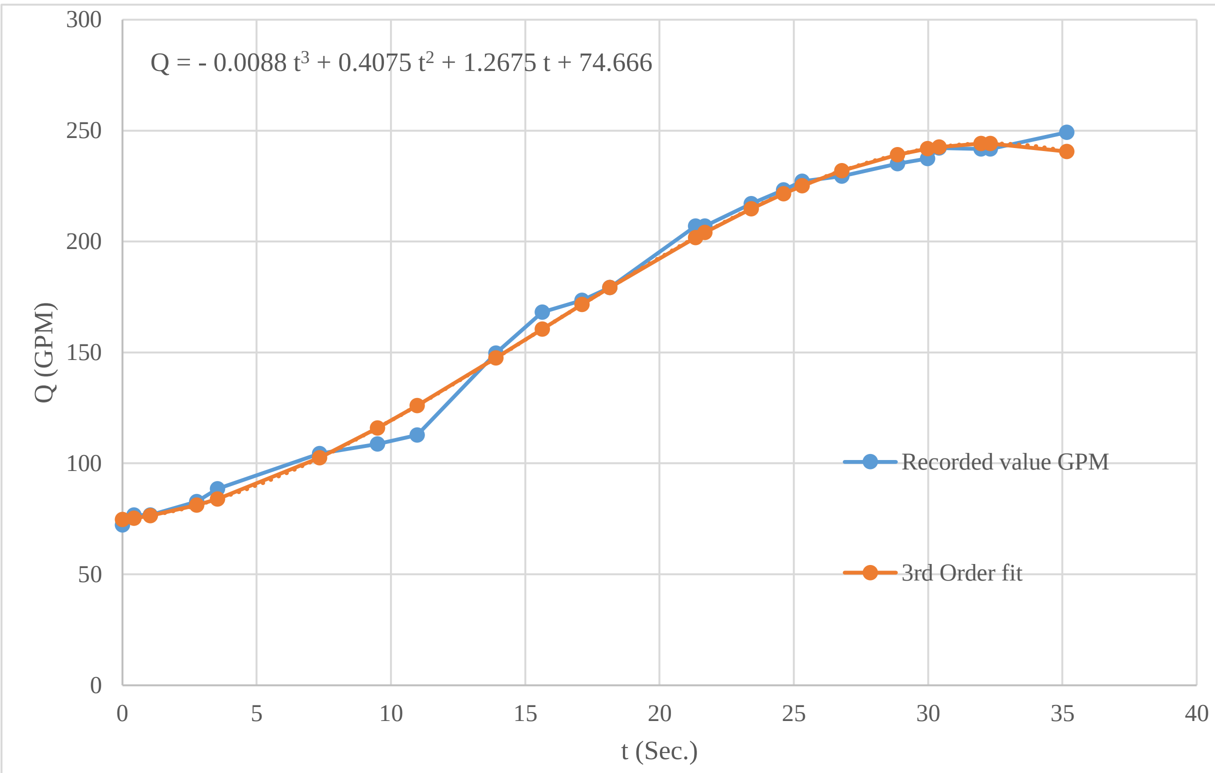


Figure G2: Increasing part of the first peak for Grundfos pumps' flow rate in Case 2, Test #2 (refer to Fig. 64 in main text for increasing peak)

Table G2: Grundfos pumps' flow rate comparison with the 3rd order equation of Fig. G2

Time Interval (sec)	Accumulated Time (sec)	Recorded Values (GPM)	3rd Order Flow Rate Fit (GPM)	Error (%)
0.432	0.000	72.207	74.666	-3.405
0.432	0.432	76.610	75.289	1.724
0.604	1.036	76.610	76.407	0.265
1.729	2.765	82.774	81.100	2.022
0.777	3.542	88.498	83.877	5.221
3.802	7.344	104.348	102.467	1.802
2.160	9.504	108.751	115.966	-6.634
1.469	10.973	112.713	126.013	-11.800
2.937	13.910	149.698	147.459	1.495
1.728	15.638	168.190	160.487	4.580
1.469	17.107	173.473	171.548	1.110
1.037	18.144	179.197	179.251	-0.030
3.197	21.341	206.935	201.775	2.493
0.345	21.686	206.935	204.046	1.396
1.728	23.414	217.061	214.785	1.049
1.210	24.624	223.225	221.572	0.741
0.691	25.315	227.188	225.136	0.903
1.469	26.784	229.389	231.862	-1.078
2.073	28.857	235.113	239.114	-1.702
1.124	29.981	237.315	241.804	-1.892
0.431	30.412	242.158	242.582	-0.175
1.556	31.968	241.717	244.137	-1.001
0.345	32.313	241.717	244.203	-1.028
2.852	35.165	249.202	240.482	3.499
	17.107		171.548	
	35.165		240.482	
	7.344		102.467	

The increasing part of the first peak of Grundfos pumps' flow rate lasted for 35.165 sec (see Fig. G2). Therefore, three points in time were selected in order to determine the Grundfos pumps' efficiency. These points in time were 7.344 sec, 17.107 sec, and 35.165 sec. These points were the same points taken for the power consumption curve fit (Table G.1) in order to have exactly the same operating points in time. Thus, the Grundfos pumps' efficiency could be determined

Discharge Pressure Curve Fitting

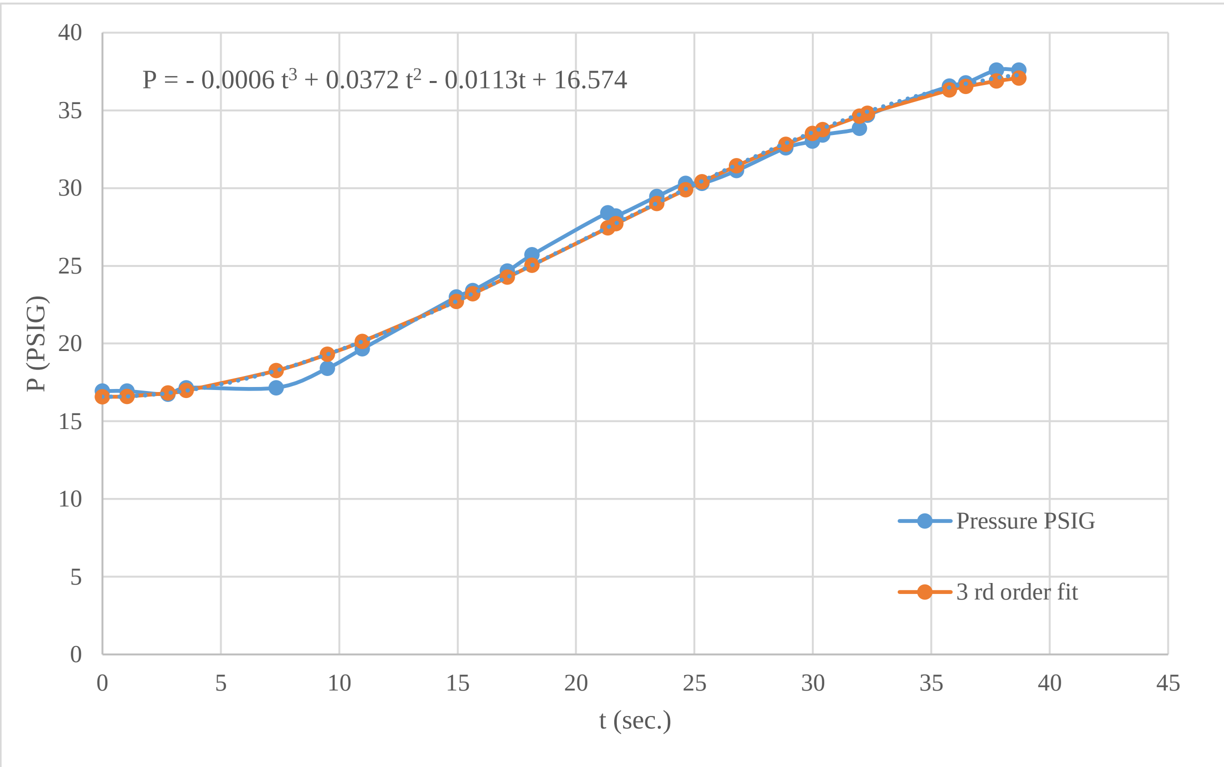


Figure G3: Increasing part of the first peak for discharge pressure of Grundfos pumps in Case 2, Test #2 (refer to Fig. 65 in main text for increasing peak)

Table G3: Grundfos pumps' discharge pressure comparison with the 3rd order equation of Fig. G3

Time Interval (sec)	Accumulated Time (sec)	Recorded Values (PSIG)	3rd Order Discharge Pressure Fit (PSIG)	Error (%)
0.000	0.000	16.940	16.574	2.163
1.036	1.036	16.940	16.602	2.000
1.729	2.765	16.737	16.814	-0.461
0.777	3.542	17.158	16.974	1.072
3.802	7.344	17.158	18.260	-6.421
2.160	9.504	18.405	19.312	-4.925
1.469	10.973	19.653	20.136	-2.462
3.974	14.947	22.988	22.712	1.201
0.691	15.638	23.409	23.200	0.894
1.469	17.107	24.656	24.263	1.594
1.037	18.144	25.701	25.032	2.604
3.197	21.341	28.413	27.443	3.412
0.345	21.686	28.210	27.704	1.792
1.728	23.414	29.457	29.001	1.547
1.210	24.624	30.298	29.893	1.337
0.691	25.315	30.298	30.394	-0.315
1.469	26.784	31.125	31.429	-0.978
2.073	28.857	32.590	32.807	-0.667
1.124	29.981	33.011	33.504	-1.493
0.431	30.412	33.417	33.760	-1.026
1.556	31.968	33.837	34.628	-2.335
0.345	32.313	34.679	34.807	-0.371
3.456	35.769	36.550	36.306	0.666
0.691	36.460	36.753	36.533	0.598
1.296	37.756	37.594	36.883	1.889
0.951	38.707	37.594	37.076	1.378
	7.344		18.260	
	17.107		24.263	
	35.165		36.087	

The increasing part of the first peak of the discharge pressure curve lasted for 38.707 sec (see Fig. G3). Therefore, three points in time were selected in order to determine the Grundfos pumps' efficiency. These points in time were 7.344 sec, 17.107 sec, and 35.165 sec. These points in time were the same points taken for the power consumption (Table G.1) and flow rate curve fit (Table G.2) in order to have exactly the same operating points in time. Thus, the Grundfos pumps' efficiency could be determined.

Appendix H: Energy Consumption Calculations

Case 2 Energy Consumption Calculations

Estimated energy consumption calculations for the day on which Test#1 was performed, March 11, 2015. The total steam generated was 709,900 lb.

For Grundfos Pumps:

Average power consumption = 1.592 kW (from Test #1, March 11)

Estimated average power consumption for March 11, 2015

$$E_p = 1.592 \text{ kW } (24 \text{ hr}) = 38.219 \text{ kW-hr}$$

For Worthington Pump:

Average power consumption = 3.999 kW (from Test #1, March 11)

Estimated average power consumption for March 11, 2015

$$= 3.999 \text{ kW } (24 \text{ hr}) = 95.989 \text{ kW-hr}$$

Estimated average energy consumption for the month of March, 2015 for Grundfos and Worthington pumps is shown in the Table H.1.

The daily energy consumption in Table H.1 was calculated using Eq. (20).

Table H1: Estimated average daily energy consumption in kW-hr

Mar 2015	TOTAL STEAM GENERATED (lb)	Grundfos Pumps (kW-hr)	Worthington Pump (kW-hr)
1	1368500	73.676	185.043
2	1290700	69.488	174.523
3	1163400	62.634	157.310
4	1379900	74.290	186.584
5	1453400	78.247	196.523
6	1121300	60.368	151.617
7	873700	47.038	118.138
8	799700	43.054	108.132
9	860200	46.311	116.313
10	824300	44.378	111.458
11	709900	*38.219	*95.990
12	643600	34.650	87.025
13	624800	33.638	84.483
14	691700	37.239	93.529
15	683400	36.792	92.406
16	577900	31.113	78.141
17	759500	40.889	102.696
18	934600	50.316	126.373
19	977400	52.621	132.160
20	854900	46.026	115.596
21	692600	37.288	93.650
22	646500	34.806	87.417
23	771300	41.525	104.292
24	867600	46.709	117.313
25	878500	47.296	118.787
26	949400	51.113	128.374
27	1013300	54.553	137.014
28	889000	47.861	120.207
29	772600	41.595	104.468
30	724900	39.027	98.018
31	619900	33.374	83.820
Total month estimated energy consumption kW-hr =		1476.134	3707.40

***** Known/ inputs for calculations in Eq. (20)

1- The energy consumption of other cold months, i.e., November through February, was based on the estimated energy calculated for March, 2015.

2- The energy consumption of other hot months, i.e., May through October, was based on the estimated energy calculated for April, 2015.

From Table H1:

Estimated energy consumption of March, 2015 of Worthington pump = 3,707.4 kW-hr

Estimated energy consumption of March, 2015 of Grundfos pumps = 1,476.1 kW-h

Estimated average energy consumption of April, 2015 of Grundfos pumps (Sections 3.2.2, 3.2.3, 3.2.4)

$\dot{W}$ = 1.1609 kW (Test #2)

$\dot{W}$ = 1.619 kW (Test #3)

$\dot{W}$ = 1 kW (Test #4)

$\dot{W}$ = Estimated average of Test #s 2, 3, and 4 for April, 2015 of Grundfos pumps = 1.263 kW

All of the tests above were performed in April of 2015. Because this thesis was written in mid-April of 2015, the total steam generated in April, 2015 was not available. Therefore, in order to calculate the other hot months (May through October), the total steam generated in April, 2014 was used.

E_p = Estimated average energy consumption in April, 2015

= (Power consumption in kW) (30 Days) (24 hr/Day) = 909.4 kW-hr

Estimated average energy consumption of April, 2015 of Worthington pump (Sections 3.2.2, 3.2.3, 3.2.4)

$\dot{W}$ = 3.993 kW (Test #2)

$\dot{W}$ = 3.934 kW (Test #3)

$\dot{W}$ = 3.995 kW (Test #4)

$\dot{W}$ = Estimated average of Test #s 2, 3, and 4 for April, 2015 of Worthington pump = 3.974 kW

E_p= Estimated average energy consumption in April, 2015

= (Power consumption in kW) (30 Day) (24 hr/Day) = 2,956.6 kW-hr

Equations (H1.1) and (H1.2) were used in order to calculate the annual energy consumption.

$$E_p\ \Big(\ Hot\ months\ (Summer)\Big) = E_p\big(April,\ 2015\big)\frac{S_G\ \big(Desired\ Month\big)\ lb}{S_G\ \big(April\ 2014\big)\ \ lb} \qquad (H1.1)$$

$$E_p\ \Big(\ Cold\ months\ (Winter)\Big) = E_p(March, 2015)\frac{S_G\ \big(Desired\ Month\big)\ lb}{S_G\ \big(March\ 2015\big)\ \ lb} \qquad (H1.2)$$

Notice that the number of days in each month cancel out as shown below. Therefore, the number of days in a month did not affect the calculations (See Eqs. (H1.1a) and (H1.1b)). For example, in order to calculate the estimated energy consumption for the month of May, 2014.

$$E_p\ \big(\ May, 2014\big) =\ E_p\ (April, 2015)\left(\frac{31\ days}{30\ days}\right)\frac{\dfrac{S_G\ \ (\ May, 2014)lb}{31\ days}}{\dfrac{S_G\ (April, 2014)lb}{30\ days}} \qquad (H1.1a)$$

or:

$$E_p\ (May\ , 2014) = E_p\ \left(April,\ \ 2015\right) \frac{S_G\ (May, 2014) lb}{S_G\ \left(April,\ 2014\right) lb} \qquad (H1.1b)$$

Based on the equations (H1.1) and (H1.2), the annual energy consumption was calculated and shown in Table H2.

Steam generated in April, 2014 = 20,085,100 lb

Steam generated in March, 2015 = 27,418,400 lb

Again, the steam generated in April, 2014 was used to calculate the energy consumption of the hot months (May to October) because the total steam generated in April, 2015 was not available at the time when this thesis was written.

Table H2: Estimated average annual energy consumption of Worthington and Grundfos pumps

Month	Year	Total Steam Generated (lb)	Grundfos Pumps (kW-hr)	Worthington Pump (kW-hr)
April	2014	20,085,100	909.44*	2,956.65*
May	2014	13,838,600	626.60	2,037.13
June	2014	13,903,500	629.54	2,046.68
July	2014	13,015,900	589.35	1,916.02
August	2014	13,637,200	617.48	2,007.48
September	2014	14,175,800	641.87	2,086.77
October	2014	17,721,800	802.43	2,608.76
November	2014	31,515,000	1,696.68	4,261.33
December	2014	35,417,400	1,906.78	4,788.99
January	2015	37,821,700	2,036.22	5,114.09
February	2015	38,179,100	2,055.46	5,162.42
March	2015	27,418,400	1,476.13*	3,707.4*
Total estimated energy consumption			**13,987.98**	**38,693.71**

* Known/ input values for calculations of Eqs. (H1.1) and (H1.2)

Appendix I: Pumps LCCA (Comparative Analysis)

NIST BLCC 5.3-13: Comparative Analysis

Consistent with Federal Life Cycle Cost Methodology and Procedures, 10 CFR, Part 436, Subpart A

Base Case: Worthington D-824 Constant Speed Pump

Alternative: Grundfos CRE 15-3 pumps

General Information

File Name:	E:\Graduation Work\Thesis\Result\Pumps LCCA\With Considering the month number of days\LCCA Programe.xml
Date of Study:	Tue May 05 15:33:54 CDT 2015
Project Name:	Pumps LCCA comparison
Project Location:	Kansas
Analysis Type:	FEMP Analysis, Energy Project
Analyst:	Raoof Alabdullah
Comment	Comparison between Grundfos CRE 15-3 pumps aginst Worthington D-824 pump
Base Date:	April 1, 2014
Service Date:	April 1, 2014
Study Period:	20 years 0 months(April 1, 2014 through March 31, 2034)
Discount Rate:	3%
Discounting Convention:	End-of-Year

Comparison of Present-Value Costs

PV Life-Cycle Cost

	Base Case	Alternative	Savings from Alternative
Initial Investment Costs:			
Capital Requirements as of Base Date	$15,472	$17,972	-$2,500
Future Costs:			
Energy Consumption Costs	$44,891	$16,228	$28,663
Energy Demand Charges	$7,566	$7,566	$0
Energy Utility Rebates	$0	$0	$0
Water Costs	$412,870	$412,870	$0
Recurring and Non-Recurring OM&R Costs	$15,623	$16,367	-$744
Capital Replacements	$1,355	$1,355	$0
Residual Value at End of Study Period	$0	$0	$0
	------------	------------	------------
Subtotal (for Future Cost Items)	$482,304	$454,386	$27,918
	------------	------------	------------
Total PV Life-Cycle Cost	$497,776	$472,358	$25,418

Net Savings from Alternative Compared with Base Case

PV of Non-Investment Savings	$27,918
- Increased Total Investment	$2,500

Net Savings	$25,418

Savings-to-Investment Ratio (SIR)

SIR = 11.17

Adjusted Internal Rate of Return

AIRR = 16.21%

Payback Period

Estimated Years to Payback (from beginning of Service Period)

Simple Payback occurs in year	2
Discounted Payback occurs in year	2

Energy Savings Summary

Energy Savings Summary (in stated units)

Energy	**-----Average**	**Annual**	**Consumption-----**	**Life-Cycle**
Type	**Base Case**	**Alternative**	**Savings**	**Savings**
Electricity	38,693.7 kWh	13,988.0 kWh	24,705.7 kWh	494,047.0 kWh

Energy Savings Summary (in MBtu)

Energy	**-----Average**	**Annual**	**Consumption-----**	**Life-Cycle**
Type	**Base Case**	**Alternative**	**Savings**	**Savings**
Electricity	132.0 MBtu	47.7 MBtu	84.3 MBtu	1,685.8 MBtu

Emissions Reduction Summary

Energy	**-----Average**	**Annual**	**Emissions-----**	**Life-Cycle**
Type	**Base Case**	**Alternative**	**Reduction**	**Reduction**
Electricity				
CO2	35,747.48 kg	12,922.90 kg	22,824.58 kg	456,429.07 kg
SO2	111.73 kg	40.39 kg	71.34 kg	1,426.61 kg
NOx	73.70 kg	26.64 kg	47.06 kg	940.99 kg
Total:				
CO2	35,747.48 kg	12,922.90 kg	22,824.58 kg	456,429.07 kg
SO2	111.73 kg	40.39 kg	71.34 kg	1,426.61 kg
NOx	73.70 kg	26.64 kg	47.06 kg	940.99 kg

Cash Flow Analysis

NIST BLCC 5.3-13: Cash Flow Analysis

Consistent with Federal Life Cycle Cost Methodology and Procedures, 10 CFR, Part 436, Subpart A

General Information

File Name:	C:\Users\Raoof\Desktop\Graduation Work\Thesis\Result\Pumps LCCA\With Considering the month number of days\LCCA Programe.xml
Date of Study:	Tue Apr 28 16:21:54 PDT 2015
Analysis Type:	FEMP Analysis, Energy Project
Project Name:	Pumps LCCA comparison
Project Location:	Kansas
Analyst:	Raoof Alabdullah
Comment:	Comparison between Grundfos CRE 15-3 pumps aginst Worthington D-824 pump
Base Date:	April 1, 2014
Service Date:	April 1, 2014
Study Period:	20 years 0 months (April 1, 2014 through March 31, 2034)

End-of-year cash-flow convention used

All costs in constant dollars (excluding general inflation)

Alternative: Worthington D-824 Constant Speed Pump

Initial Capital Costs

Component:

Year Beginning	Total
Apr 2014	$12,500
Total	$12,500

Capital Investment Costs

Year Beginning	Initial	Replacement	Total
Apr 2014	$12,500	$0	$12,500
Apr 2015	$0	$0	$0
Apr 2016	$0	$200	$200
Apr 2017	$0	$0	$0
Apr 2018	$0	$200	$200
Apr 2019	$0	$0	$0
Apr 2020	$0	$200	$200
Apr 2021	$0	$0	$0
Apr 2022	$0	$200	$200
Apr 2023	$0	$0	$0
Apr 2024	$0	$200	$200

Apr 2025	$0	$0	$0
Apr 2026	$0	$200	$200
Apr 2027	$0	$0	$0
Apr 2028	$0	$200	$200
Apr 2029	$0	$0	$0
Apr 2030	$0	$200	$200
Apr 2031	$0	$0	$0
Apr 2032	$0	$200	$200
Apr 2033	$0	$0	$0
------------	------------	------------	------------
Total	$12,500	$1,800	$14,300

Operating-Related Costs

Year Beginning	Recurring	Non-Recurring	Energy Consumption	Energy Demand	Energy Rebate	Water	Water Disp.	Total
Apr 2014	$1,000	$0	$2,818	$475	$0	$27,749	$0	$32,043
Apr 2015	$1,000	$0	$2,874	$484	$0	$27,749	$0	$32,108
Apr 2016	$1,000	$0	$2,958	$499	$0	$27,749	$0	$32,206
Apr 2017	$1,000	$0	$2,998	$505	$0	$27,749	$0	$32,253
Apr 2018	$1,000	$0	$2,997	$505	$0	$27,749	$0	$32,251
Apr 2019	$1,000	$0	$2,970	$501	$0	$27,749	$0	$32,220
Apr 2020	$1,000	$0	$2,975	$501	$0	$27,749	$0	$32,226
Apr 2021	$1,000	$0	$2,991	$504	$0	$27,749	$0	$32,244
Apr 2022	$1,000	$0	$3,017	$508	$0	$27,749	$0	$32,275
Apr 2023	$1,000	$0	$3,040	$512	$0	$27,749	$0	$32,302
Apr 2024	$1,000	$1,000	$3,051	$514	$0	$27,749	$0	$33,315
Apr 2025	$1,000	$0	$3,057	$515	$0	$27,749	$0	$32,322
Apr 2026	$1,000	$0	$3,062	$516	$0	$27,749	$0	$32,327
Apr 2027	$1,000	$0	$3,068	$517	$0	$27,749	$0	$32,335
Apr 2028	$1,000	$0	$3,080	$519	$0	$27,749	$0	$32,349
Apr 2029	$1,000	$0	$3,091	$521	$0	$27,749	$0	$32,362
Apr 2030	$1,000	$0	$3,105	$523	$0	$27,749	$0	$32,378
Apr 2031	$1,000	$0	$3,130	$528	$0	$27,749	$0	$32,407
Apr 2032	$1,000	$0	$3,153	$531	$0	$27,749	$0	$32,434
Apr 2033	$1,000	$0	$3,184	$537	$0	$27,749	$0	$32,470
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Total	$20,000	$1,000	$60,622	$10,218	$0	$554,986	$0	$646,825

Sum of All Cash Flows

Year Beginning	Capital	OM&R	Total
Apr 2014	$12,500	$32,043	$44,543

Apr 2015	$0	$32,108	$32,108
Apr 2016	$200	$32,206	$32,406
Apr 2017	$0	$32,253	$32,253
Apr 2018	$200	$32,251	$32,451
Apr 2019	$0	$32,220	$32,220
Apr 2020	$200	$32,226	$32,426
Apr 2021	$0	$32,244	$32,244
Apr 2022	$200	$32,275	$32,475
Apr 2023	$0	$32,302	$32,302
Apr 2024	$200	$33,315	$33,515
Apr 2025	$0	$32,322	$32,322
Apr 2026	$200	$32,327	$32,527
Apr 2027	$0	$32,335	$32,335
Apr 2028	$200	$32,349	$32,549
Apr 2029	$0	$32,362	$32,362
Apr 2030	$200	$32,378	$32,578
Apr 2031	$0	$32,407	$32,407
Apr 2032	$200	$32,434	$32,634
Apr 2033	$0	$32,470	$32,470
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Total	$14,300	$646,825	$661,125

Alternative: Grundfos CRE 15-3 pumps

Initial Capital Costs

Component:

Year Beginning	Total
Apr 2014	$15,000
Total	$15,000

Capital Investment Costs

Year Beginning	Initial	Replacement	Total
Apr 2014	$15,000	$0	$15,000
Apr 2015	$0	$0	$0
Apr 2016	$0	$200	$200
Apr 2017	$0	$0	$0
Apr 2018	$0	$200	$200
Apr 2019	$0	$0	$0
Apr 2020	$0	$200	$200
Apr 2021	$0	$0	$0
Apr 2022	$0	$200	$200

Apr 2023	$0	$0	$0
Apr 2024	$0	$200	$200
Apr 2025	$0	$0	$0
Apr 2026	$0	$200	$200
Apr 2027	$0	$0	$0
Apr 2028	$0	$200	$200
Apr 2029	$0	$0	$0
Apr 2030	$0	$200	$200
Apr 2031	$0	$0	$0
Apr 2032	$0	$200	$200
Apr 2033	$0	$0	$0
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Total	$15,000	$1,800	$16,800

Operating-Related Costs

Year Beginning	Recurring	Non-Recurring	Energy Consumption	Energy Demand	Energy Rebate	Water	Water Disp.	Total
Apr 2014	$1,000	$0	$1,019	$475	$0	$27,749	$0	$30,243
Apr 2015	$1,000	$0	$1,039	$484	$0	$27,749	$0	$30,273
Apr 2016	$1,000	$0	$1,069	$499	$0	$27,749	$0	$30,317
Apr 2017	$1,000	$0	$1,084	$505	$0	$27,749	$0	$30,339
Apr 2018	$1,000	$0	$1,083	$505	$0	$27,749	$0	$30,338
Apr 2019	$1,000	$0	$1,074	$501	$0	$27,749	$0	$30,324
Apr 2020	$1,000	$0	$1,075	$501	$0	$27,749	$0	$30,326
Apr 2021	$1,000	$0	$1,081	$504	$0	$27,749	$0	$30,334
Apr 2022	$1,000	$0	$1,091	$508	$0	$27,749	$0	$30,348
Apr 2023	$1,000	$0	$1,099	$512	$0	$27,749	$0	$30,361
Apr 2024	$1,000	$2,000	$1,103	$514	$0	$27,749	$0	$32,366
Apr 2025	$1,000	$0	$1,105	$515	$0	$27,749	$0	$30,370
Apr 2026	$1,000	$0	$1,107	$516	$0	$27,749	$0	$30,372
Apr 2027	$1,000	$0	$1,109	$517	$0	$27,749	$0	$30,376
Apr 2028	$1,000	$0	$1,114	$519	$0	$27,749	$0	$30,382
Apr 2029	$1,000	$0	$1,118	$521	$0	$27,749	$0	$30,388
Apr 2030	$1,000	$0	$1,123	$523	$0	$27,749	$0	$30,395
Apr 2031	$1,000	$0	$1,132	$528	$0	$27,749	$0	$30,408
Apr 2032	$1,000	$0	$1,140	$531	$0	$27,749	$0	$30,421
Apr 2033	$1,000	$0	$1,151	$537	$0	$27,749	$0	$30,437
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Total	$20,000	$2,000	$21,915	$10,218	$0	$554,986	$0	$609,118

Sum of All Cash Flows

Year Beginning	Capital	OM&R	Total
Apr 2014	$15,000	$30,243	$45,243
Apr 2015	$0	$30,273	$30,273
Apr 2016	$200	$30,317	$30,517
Apr 2017	$0	$30,339	$30,339
Apr 2018	$200	$30,338	$30,538
Apr 2019	$0	$30,324	$30,324
Apr 2020	$200	$30,326	$30,526
Apr 2021	$0	$30,334	$30,334
Apr 2022	$200	$30,348	$30,548
Apr 2023	$0	$30,361	$30,361
Apr 2024	$200	$32,366	$32,566
Apr 2025	$0	$30,370	$30,370
Apr 2026	$200	$30,372	$30,572
Apr 2027	$0	$30,376	$30,376
Apr 2028	$200	$30,382	$30,582
Apr 2029	$0	$30,388	$30,388
Apr 2030	$200	$30,395	$30,595
Apr 2031	$0	$30,408	$30,408
Apr 2032	$200	$30,421	$30,621
Apr 2033	$0	$30,437	$30,437
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Total	$16,800	$609,118	$625,918

Summary LCC

NIST BLCC 5.3-13: Summary LCC

Consistent with Federal Life Cycle Cost Methodology and Procedures, 10 CFR, Part 436, Subpart A

General Information

File Name:	E:\Graduation Work\Thesis\Result\Pumps LCCA\With Considering the month number of days\LCCA Programe.xml
Date of Study:	Tue May 05 15:32:14 CDT 2015
Analysis Type:	FEMP Analysis, Energy Project
Project Name:	Pumps LCCA comparison
Project Location:	Kansas
Analyst:	Raoof Alabdullah
Comment:	Comparison between Grundfos CRE 15-3 pumps aginst Worthington D-824 pump
Base Date:	April 1, 2014
Service Date:	April 1, 2014
Study Period:	20 years 0 months (April 1, 2014 through March 31, 2034)
Discount Rate:	3%
Discounting Convention:	End-of-Year

Discount and Escalation Rates are REAL (exclusive of general inflation)

Alternative: Worthington D-824 Constant Speed Pump

LCC Summary

	Present Value	Annual Value
Initial Cost	$15,472	$1,040
Energy Consumption Costs	$44,891	$3,018
Energy Demand Costs	$7,566	$509
Energy Utility Rebates	$0	$0
Water Usage Costs	$412,870	$27,754
Water Disposal Costs	$0	$0
Annually Recurring OM&R Costs	$14,879	$1,000
Non-Annually Recurring OM&R Costs	$744	$50
Replacement Costs	$1,355	$91
Less Remaining Value	$0	$0
	------------	------------
Total Life-Cycle Cost	$497,776	$33,462

Alternative: Grundfos CRE 15-3 pumps

LCC Summary

	Present Value	Annual Value

Initial Cost	$17,972	$1,208
Energy Consumption Costs	$16,228	$1,091
Energy Demand Costs	$7,566	$509
Energy Utility Rebates	$0	$0
Water Usage Costs	$412,870	$27,754
Water Disposal Costs	$0	$0
Annually Recurring OM&R Costs	$14,879	$1,000
Non-Annually Recurring OM&R Costs	$1,488	$100
Replacement Costs	$1,355	$91
Less Remaining Value	$0	$0
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Total Life-Cycle Cost	$472,358	$31,753

Appendix J: Vent Condenser LCCA

Vent Condenser LCCA (Summary LCC)

NIST BLCC 5.3-13: Summary LCC

Consistent with Federal Life Cycle Cost Methodology and Procedures, 10 CFR, Part 436, Subpart A

General Information

File Name:	E:\Graduation Work\Thesis\Figures\DATA\Result\Vent Calculation\Vent Condenser LCCA Project.xml
Date of Study:	Wed Apr 22 18:00:09 CDT 2015
Analysis Type:	FEMP Analysis, Energy Project
Project Name:	Vent Condenser Reclaim Energy
Project Location:	Kansas
Analyst:	Raoof Alabdullah
Comment:	The vent condenser saving natural gas analysis
Base Date:	April 1, 2014
Service Date:	April 1, 2014
Study Period:	20 years 0 months (April 1, 2014 through March 31, 2034)
Discount Rate:	3%
Discounting Convention:	End-of-Year

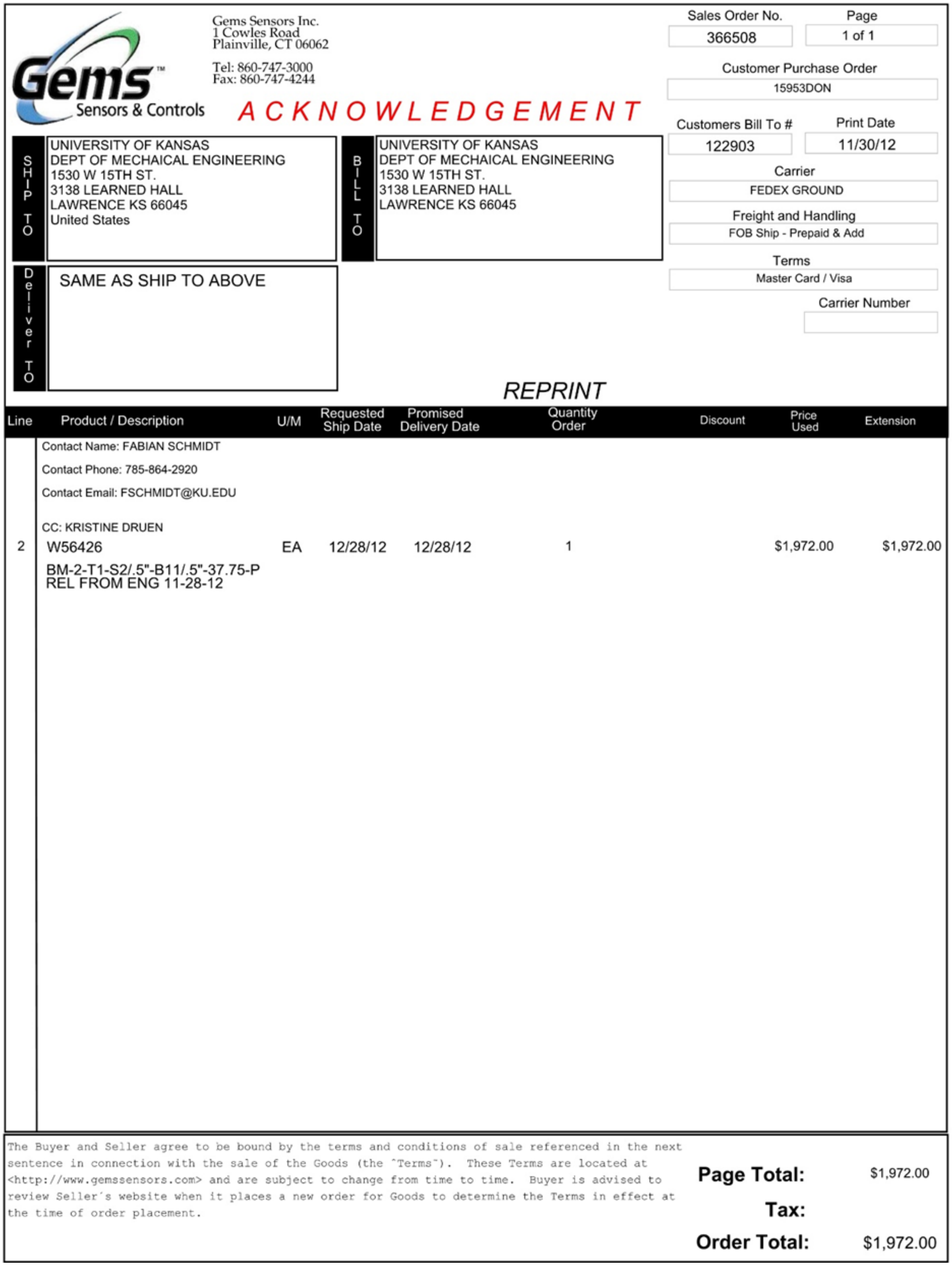
Discount and Escalation Rates are REAL (exclusive of general inflation)

Alternative: Vent Condenser Analysis

LCC Summary

	Present Value	Annual Value
Initial Cost	$10,000	$672
Energy Consumption Costs	$846,925	$56,932
Energy Demand Costs	$0	$0
Energy Utility Rebates	$0	$0
Water Usage Costs	$0	$0
Water Disposal Costs	$0	$0
Annually Recurring OM&R Costs	$0	$0
Non-Annually Recurring OM&R Costs	$16,937	$1,139
Replacement Costs	$7,441	$500
Less Remaining Value	$0	$0
	------------	------------
Total Life-Cycle Cost	$881,303	$59,243

Appendix K: Level Sensor Invoice



Gems Sensors Inc.
1 Cowles Road
Plainville, CT 06062

Tel: 860-747-3000
Fax: 860-747-4244

Gems™ Sensors & Controls

ACKNOWLEDGEMENT

Sales Order No.	Page
366508	1 of 1

Customer Purchase Order: 15953DON

Customers Bill To #	Print Date
122903	11/30/12

Carrier: FEDEX GROUND

Freight and Handling: FOB Ship - Prepaid & Add

Terms: Master Card / Visa

Carrier Number:

SHIP TO:
UNIVERSITY OF KANSAS
DEPT OF MECHAICAL ENGINEERING
1530 W 15TH ST.
3138 LEARNED HALL
LAWRENCE KS 66045
United States

BILL TO:
UNIVERSITY OF KANSAS
DEPT OF MECHAICAL ENGINEERING
1530 W 15TH ST.
3138 LEARNED HALL
LAWRENCE KS 66045

Deliver TO:
SAME AS SHIP TO ABOVE

REPRINT

Line	Product / Description	U/M	Requested Ship Date	Promised Delivery Date	Quantity Order	Discount	Price Used	Extension
	Contact Name: FABIAN SCHMIDT Contact Phone: 785-864-2920 Contact Email: FSCHMIDT@KU.EDU CC: KRISTINE DRUEN							
2	W56426 BM-2-T1-S2/.5"-B11/.5"-37.75-P REL FROM ENG 11-28-12	EA	12/28/12	12/28/12	1		$1,972.00	$1,972.00

The Buyer and Seller agree to be bound by the terms and conditions of sale referenced in the next sentence in connection with the sale of the Goods (the "Terms"). These Terms are located at <http://www.gemssensors.com> and are subject to change from time to time. Buyer is advised to review Seller's website when it places a new order for Goods to determine the Terms in effect at the time of order placement.

Page Total: $1,972.00

Tax:

Order Total: $1,972.00